The NOVEMBER, 1931

Heating · Piping and Air Conditioning

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By A. W. Loomis

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By Arthur McCutchan

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PIPING HIGH, INTERMEDIATE AND LOW PRESSURE STEAM ECONOMICALLY
By H. L. Colby *and* Francis A. Westbrook

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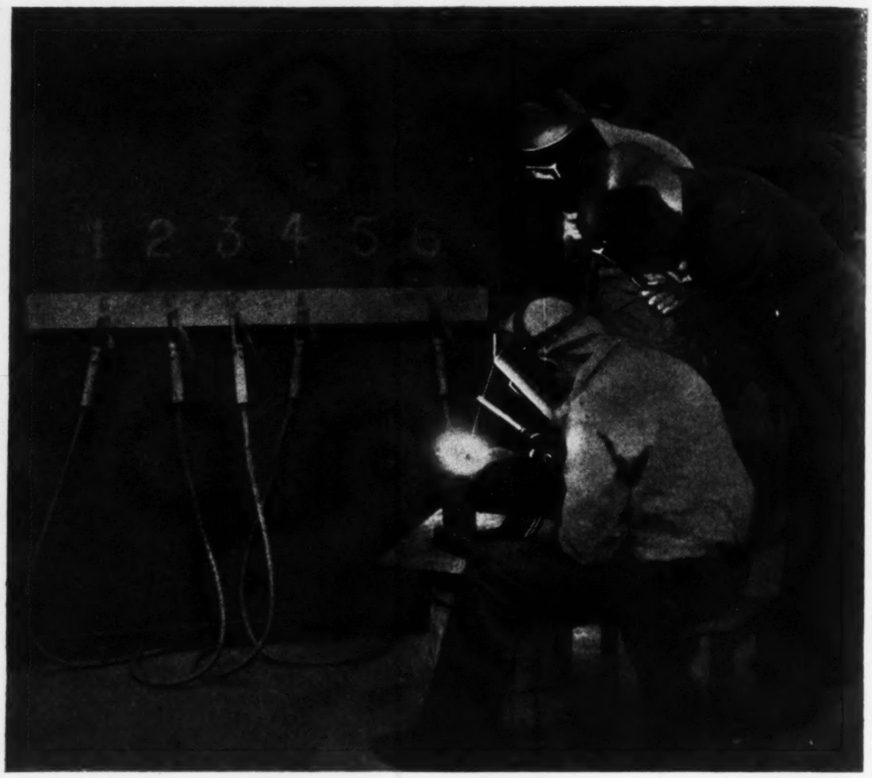
Journal Section of the A. S. H. V. E.

"THE PROOF OF THE PUDDING"



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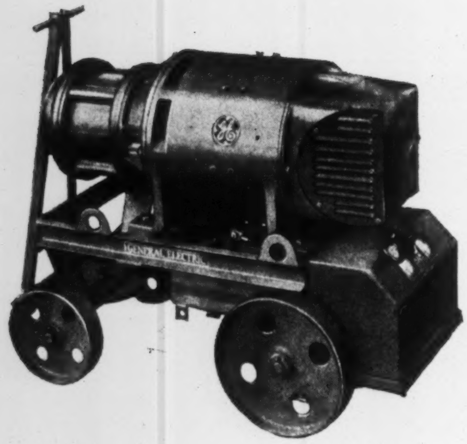

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GENERAL ELECTRIC

SALES AND ENGINEERING SERVICE IN PRINCIPAL CITIES

The EDITOR'S PAGE

Heating · Piping and Air Conditioning

FROM ARTICLES IN THIS ISSUE

Heat Loss

"To the computed heat loss of the balcony areas, 20 per cent was added arbitrarily for diffusion losses to the arena space. Subsequent performance has shown this to be more than the actual excess requirements."—*Louis L. Narowetz Jr. and R. M. Moffitt.*

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"It is interesting to know that this ash conveyor system is operated at will during office hours without the slightest hint of such activity being noticeable in even those public and working spaces through which the conveyor passes."—*Henry G. Schaefer.*

Metering Customers

"The quantity of heat supplied a customer is metered by instruments which measure the quantity of water flowing through the coils of the heat exchanger, and also the drop in temperature; these two quantities are multiplied and integrated mechanically so that the meter reading is in Btu-hours."—*Ludwig Wahl, Paul Reschke, and F. E. Giesecke.*

Necessity of Cooling

"In the light of the evidence presented here, a campaign against seasonal diseases should include, in addition to the control of the food supply and general sanitation work, the control of the indoor air conditions, particularly in institutions for sick infants and children. Protection against excessive summer heat is still a rare exception in institutions of this kind, but the number of new hospitals providing for artificially cooled rooms for infants is gradually increasing."—*C. P. Yaglou*

Saving Power

"The difference in cost between installing 12-in. and 10-in. pipe for these sections (of air piping in a mine), is $5,820. The power saving using 12-in. pipe would, therefore, pay for the larger pipe in less than 3½ years. This is ample justification for choosing the larger pipe."—*A. W. Loomis.*

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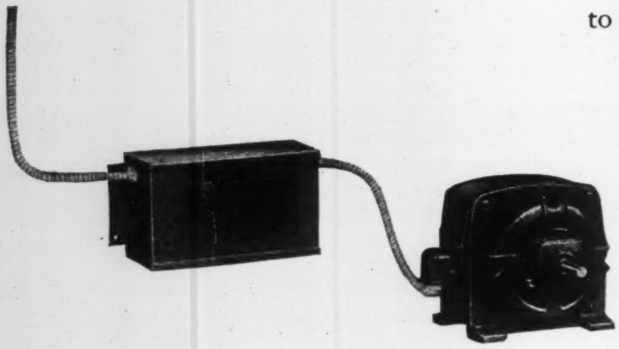

G-E capacitor-motor connected to capacitor-transformer unit

GENERAL ELECTRIC

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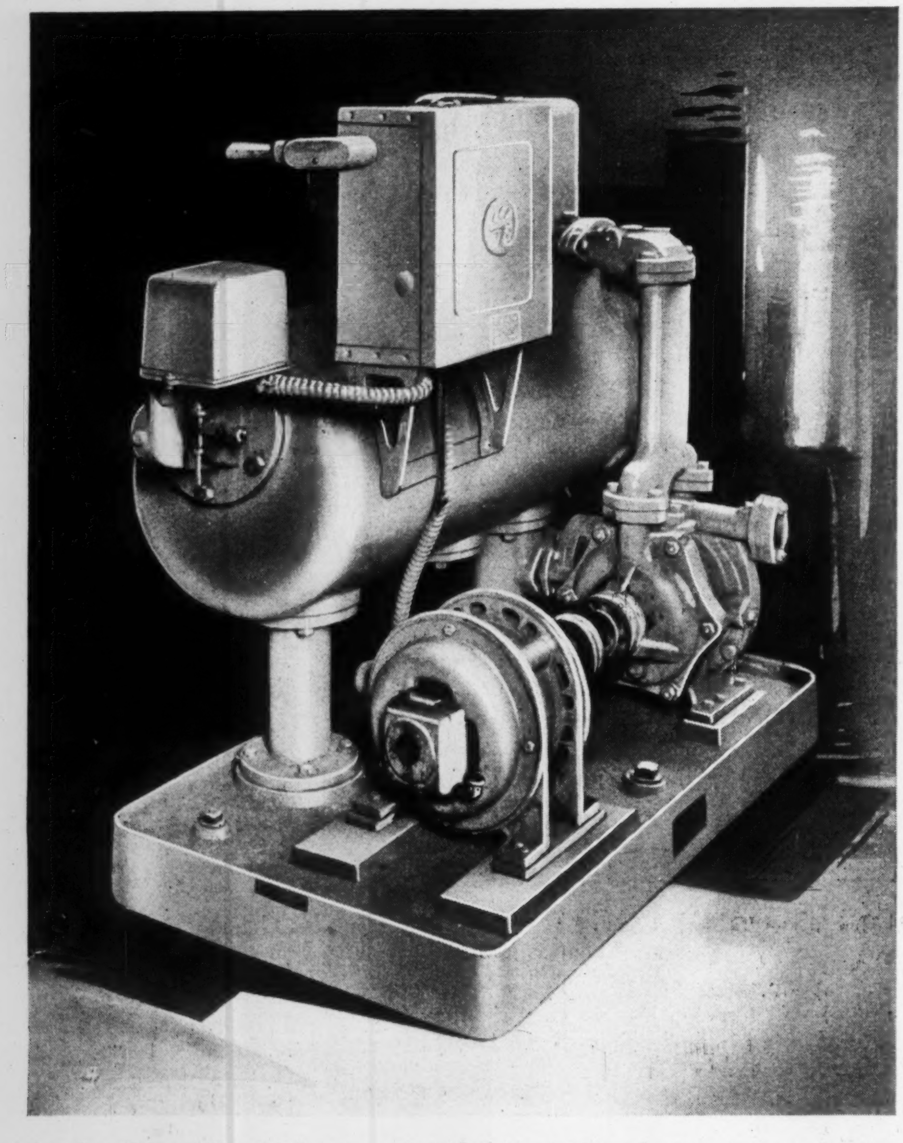

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Heating · Piping
and Air Conditioning

Vol. III November, 1931 No. 11

Piping Analysis Assures Minimum Compressed Air Cost

By A. W. Loomis*

This article tells how the air piping system for a large mine was relocated and modernized. By careful study and proper procedure in planning air distributing systems, minimum power and maintenance costs are assured. While there are problems peculiar to mines described in this paper, the article may well serve as an explanation of how to improve air piping in industrial plants of any type.

CONSTRUCTION of a new tunnel connecting the various workings of a large mine made it necessary to relocate entirely the air piping and to modernize the air compressor system.

The old pipes were run down slope shafts which followed the angle of the lode. They were in poor condition, and, as the ground was very shifty, maintenance was expensive and difficult. It was planned to abandon these slope shafts as soon as the pillars surrounding them had been robbed of their ore. As soon as the work of removing the pillars was started, the pipes in these shafts would be inaccessible for maintenance.

Determining Required Air Quantities

It was necessary, of course, to find first the quantity of air needed. It was calculated by three separate methods: (1) An estimate of the total air consumption of drills and other mining equipment which was to be operated; (2) a check on the past operating records of the compressors; (3) a check on the past usage per ton of ore produced.

Figures of 10,000, 12,000 and 12,700 cu. ft. of free air per min. actual delivery were obtained by these methods. It was, therefore, thought that 12,000 c.f.m. was a reasonable figure on which to base calculations. However, to provide for future growth, emergency use, and a reasonable load factor, this was raised to 15,000 c.f.m.

These figures were based on a discharge pressure of 85 lb. per sq. in. at the underground distributing centers. Under the old system, this pressure varied from 55 to 65 lb. per sq. in., depending on the location and the load on the air lines. The higher discharge pressure was decided upon to secure fast and efficient operation of the tools.

It was found by test that about 20 per cent of the total required capacity was leakage. It was felt that

Fig. 1—"Stopehammer" Drill in a Stope in a Mine. This Is One of the Air-Operated Tools Which the Air Piping System Described in This Article Serves

the new piping system with its greater accessibility and ease of maintenance would reduce this figure to 15 per cent, which is by no means unusual for a system of this extent.

---

*Ingersoll-Rand Company, New York City.

The factors and data which governed the arrangement of the system were as follows:

> Required ultimate capacity, 15,000 c. f. m., to be divided equally among three separate distribution points; i. e., 5000 c. f. m. each. (These points are shown as *A, B,* and *C* in Fig. 2.)
>
> Two shafts (No. 1 and No. 2 in Fig. 2) were available for pipe lines from the surface of the ground to the lower workings of the mine.
>
> Costs of pipe installed were available from previous work done at the mine and were as follows:

PIPE SIZE, IN.	LAID COST PER FOOT		
	HORIZONTAL OR ON SLOPE	VERTICAL	PER PAIR OF FLANGES
5	$1.50	....	....
6	1.80	$2.20	$4.00
8	2.35	2.80	5.00
10	2.90	3.40	6.90
12	3.50	4.00	9.65
14	4.00	4.60	13.00

With these facts as a basis, the pipe system was then sketched out.

Sizing the Vertical Piping

It was decided to put the vertical pipes down both No. 1 and No. 2 shafts because (1) it was necessary to distribute the air to widely separated points, and it was considered wiser to distribute as directly as possible and thus eliminate large pipes in the working passages; (2) two distribution lines, when properly connected by tie-in lines, would always permit operation if anything should happen to one of the lines. Such operation would, however, be at the expense of reduced pressure due to higher friction losses; (3) space in the shafts was limited and the large pipe size necessary to transmit the total amount of air would greatly restrict one of the shafts. The air quantity for each of the vertical shafts, was, therefore, indicated as 7,500 c.f.m.

Laying Out Underground Piping

The underground piping to the points of distribution (points *A, B,* and *C* in Fig. 2) was limited in its placement by the position of the drifts and stopes. The piping was laid out so that the air could reach each distribution point from either one of the vertical shafts. This, it will be noticed, was accomplished for all three distribution points. Insofar as possible, it was also arranged so that each distribution point should be located on a loop so that air could reach it from two or more directions. This was accomplished except for point *A,* which is located approximately 3,000 ft. from the tie-in lines.

With the piping arrangement completely laid out, it was a simple matter to indicate the air quantities which should be handled by each section of pipe. In the case of points *B* and *C,* which were located on a loop, the air quantities were so proportioned that about 75 per cent of the amount used came through one side of the loop and the remaining 25 per cent came through the other side of the loop. The division in this manner assured an economical installation cost and, at the same time, provided tie-in pipes of sufficient size to provide for reasonable emergency operation. This point will be discussed in more detail later.

Computing the Pipe Sizes

After the air quantities had been determined, it was then necessary to compute the necessary pipe sizes, and the cost of installation. To do this, a certain size pipe must be assumed and the loss of pressure through this pipe figured by means of an air flow formula or by tables prepared from such a formula. Tables are, of course, more convenient and are time savers. Such tables appeared in an article by the author in the March, 1930, HEATING, PIPING AND AIR CONDITIONING.

The computations which follow will be based on those tables. These tables show only the pressure drop for quantities of 5,000 c.f.m. or under. In order to simplify our sample computation, we shall, therefore, use section No. 4 as an example since the air quantity for this section falls within the limits of the tables. The

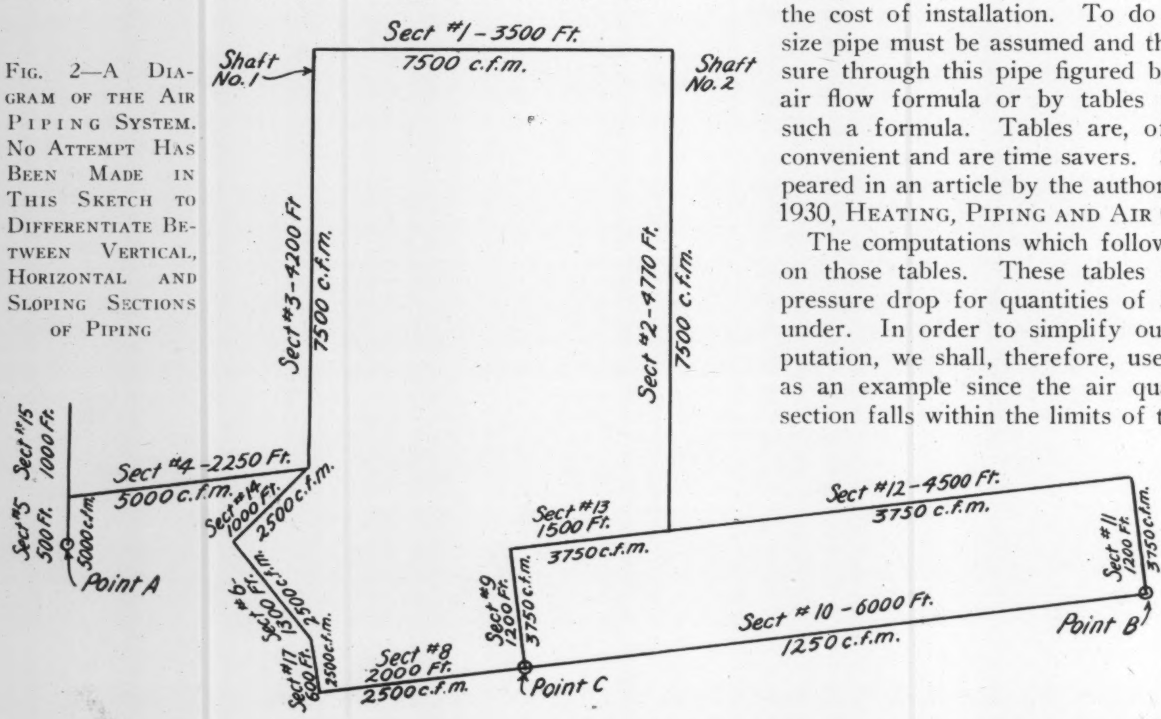

FIG. 2—A DIAGRAM OF THE AIR PIPING SYSTEM. NO ATTEMPT HAS BEEN MADE IN THIS SKETCH TO DIFFERENTIATE BETWEEN VERTICAL, HORIZONTAL AND SLOPING SECTIONS OF PIPING

larger air quantities may be computed from these tables, however, if one bears in mind that the friction in a given size of pipe will vary directly as the square of the air quantity.

Pressure Drop

Taking section No. 4 with an air quantity of 5,000 c.f.m. of free air and a length of 2,250 ft. as an example, the tables show pressure drop for 80 lb. discharge pressure in an 8-in. pipe to be 2.46 lb. per 1,000 ft. of pipe. For 2,250 feet of pipe, the pressure drop is:

$$\frac{2250}{1000} \times 2.46 = 5.53 \text{ lb.}$$

This is based on a pressure of 80 lb. per sq. in., but since the pressure desired at the distribution points is 85 lb. per sq. in., the figures from the tables should be corrected for this pressure difference. However, as will be shown later, the actual pressure in some of the pipes will be less than 85 lb. per sq. in., so the friction loss based on an average 85 lb. is near enough for practical calculations.

Correcting Tables for Other Discharge Pressures

The method of arriving at the pressure loss for 85 lb. pressure is as follows:

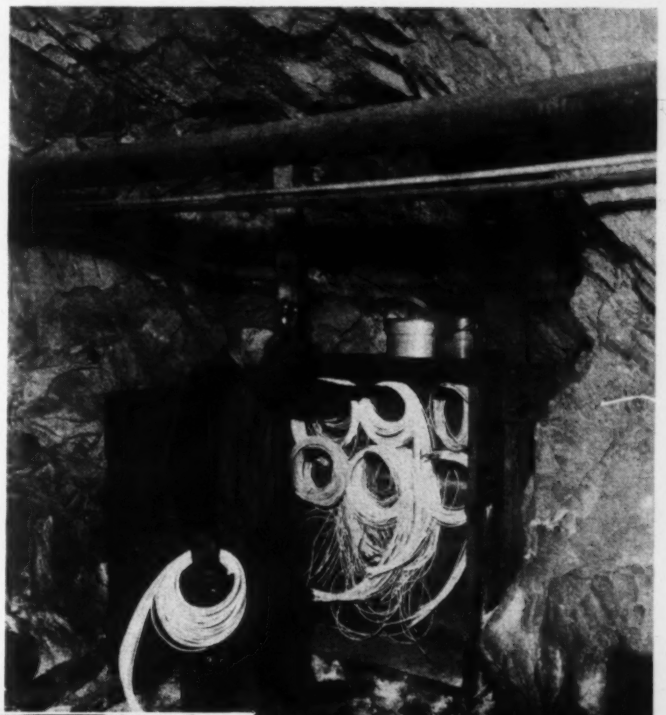

Fig. 4 (Above)—Air and Water Piping Passing Over a Fuse Storage Box in a Mine

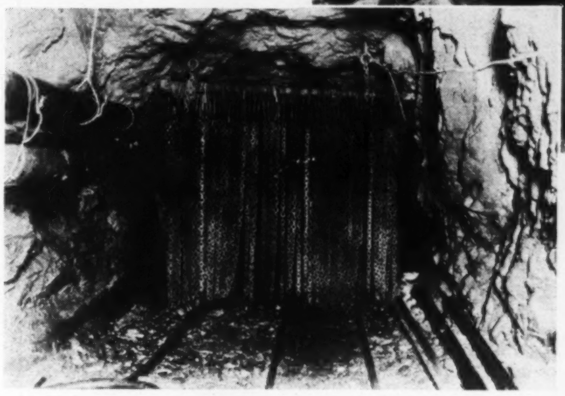

Fig. 3 — Chain Curtain Used to Stop Pieces of Flying Rock When Blasting Is in Progress. Air and Water Headers Can Be Seen at the Left of the Tunnel

The pressure drop varies inversely as the absolute pressures. The atmospheric pressure at this job varies from about 14.4 lb. per sq. in. absolute at the surface to about 17.7 lb. per sq. in. absolute in the lower workings. We may, therefore, assume 16 lb. per sq. in. as a fair average pressure. Since absolute pressure is gage plus atmospheric pressure, the pressure drop for 85 lb. per sq. in. is as follows:

$$\frac{(80+16)}{(85+16)} \times 5.53 = 5.3 \text{ lb. pressure drop in section No. 4 using 8-in. pipe.}$$

This procedure is followed for each section and the results, including the installed cost of the pipe tabulated, are shown in Table 1.

In a few sections pipe from the old system was already installed and this was used insofar as it was possible.

Choice of Correct Pipe Size an Economic Proposition

The choice of the correct pipe size is an economic proposition. If the larger size pipe is selected, the saving in power cost, due to the smaller friction loss, must pay for the increased cost of installation.

After a little experience in figuring economic pipe sizes, the choice can usually be made at sight. In this case, the larger pipe was chosen for each section and then checked by figuring the saving for sections No. 2 and No. 3. Since these sections showed the least difference in pressure drop between the smaller pipe and the larger pipe, they automatically act as a check on the choices for the other sections.

Determining Pressure for Compressors

It was necessary first of all to find the pressure at which the compressor would be required to operate.

Two factors determine this pressure; the pressure gain due to depth, and the pressure loss due to friction.

Distribution points *B* and *C* are about 4,900 feet below the surface. Assuming that the pressure at the compressor to be about 80 lb. per sq. in. and the average air temperature in the pipes 150 F, the pressure gain to distribution points *B* and *C* is 14 lb. This figure is arrived at as follows: The weight of a cubic foot of air at 80 lb. pressure and 150 F at the altitude existing is about 0.41 lb. The weight of the air column from the compressor to the distribution points is therefore:

$$\frac{0.41}{144} \times 4900 = 14 \text{ lb.}$$

Distribution point *A* is about 3,600 ft. below the surface, and the pressure gain at this point is 10.3 lb.

The pressure loss for each distribution system may be determined by adding the pressure losses in the various sections of pipe leading to that point. Since points *B* and *C* receive air from two directions, the pressure loss in both systems must be determined.

Table 1

Section of Pipe Line	Size in In.	Length in Ft.	Air Quantity in C.F.M.	Pressure Drop in Lb. per Sq. In.	Installed Cost, including Pipe Flanges
No. 2	10	4770	7500	7.8	$16,750
	12	4770	7500	3.2	19,870
No. 3	10	4200	7500	6.9	14,780
	12	4200	7500	2.7	17,480
No. 4	8	2250	5000	5.3	5,400
	10	2250	5000	1.6	6,660
No. 5	2-6	500	5000	1.3	In
No. 6	6	1300	2500	3.3	....
	8	1300	2500	0.7	3,120
No. 7	6	600	2500	1.5	1,100
	8	600	2500	0.3	1,440
No. 8	6	2000	2500	5.0	3,680
	8	2000	2500	1.1	4,800
No. 9	8	1200	3750	1.6	In
No. 10	6	6000	1250	3.5	11,040
	8	6000	1250	0.8	14,400
No. 11	6	1200	3750	6.8	In
	8	1200	3750	1.6	2,880
No. 12	8	4500	3750	5.9	10,830
	10	4500	3750	1.8	13,360
No. 13	10	1500	3750	0.6	In
No. 14	6	1000	2500	2.5	....
	8	1000	2500	0.6	2,400
No. 15	2-6	1000	?	....	In

The losses to the distribution points are shown in Table 2.

It may be assumed that the average pressure drop for the two systems serving each distribution point will be the actual pressure drop. The air quantities will change slightly to balance the friction loss, but, as the change is slight and the pressure drop is approximate at the best, no allowance need be made for this.

We now have the desired pressure at the distribution points, the friction loss and the pressure gain due to the weight of air. We may, therefore, find the pressure which the air compressors must maintain. This is shown in Table 3.

Cost of Compressed Air

The compressors run for sixteen hours a day, 310 days a year. The total piston displacement at 15,000 c.f.m. is, therefore, 4,464,000,000 cu. ft. of air per year.

Electric current costs 0.6c per kw.-hr. delivered at the mine.

Computed cost of compressed air is as follows:

Pressure	Cost of Power per Million cu. ft.	Cost of Labor and Supplies per Million Cu. Ft.	Total Cost per Million Cu. Ft.
78-lb.	$12.80	$3.00	$15.80
82 lb.	13.22	3.00	16.22

The cost of air when the compressors are operating against 82.0-lb. discharge pressure is about $72,300 a year. Against 78.3 lb., the cost is $70,500 a year, or a saving of $1,800 a year. The difference in cost between 12-in. and 10-in. on sections 2 and 3 is $5,820. The power saving in using 12-in. pipe would, therefore, pay for the larger pipe in less than 3½ years. This is ample justification for choosing the larger pipe.

Installing Compressors

Up until the present time section No. 1 of the piping system has not been considered. The pipe size in this

Table 2—Friction Loss to Distribution Points

POINT B						POINT C						POINT A		
No. of Sect.	Size In.	Fric. Loss Lb.	No. of Sect.	Size In.	Fric. Loss Lb.	No. of Sect.	Size In.	Fric. Loss Lb.	No. of Sect.	Size In.	Fric. Loss Lb.	No. of Sect.	Size In.	Fric. Loss Lb.
2	10	7.8	3	10	6.9	2	10	7.8	3	10	6.9	3	10	6.9
12	10	1.8	14	8	0.6	13	10	0.6	14	8	0.6	4	10	1.6
11	8	1.6	6	8	0.7	9	8	1.6	6	8	0.7	5	2-6	1.3
....	....	....	7	8	0.3	....	....	....	7	8	0.3	....	....	....
....	....	....	8	8	1.1	....	....	....	8	8	1.1	....	....	....
....	....	....	10	8	0.8	....	....	....	....	....	....	....	....	....
Total with 10-in. pipe at Section 2—11.2 lb. Total with 12-in. pipe at Section 3—6.6 lb.			Total with 10-in. pipe at Section 3—10.4 lb. Total with 12-in. pipe at Section 3—6.2 lb.			Total with 10-in. pipe at Section 2—10.0 lb. Total with 12-in. pipe at Section 2—5.4 lb.			Total with 10-in. pipe at Section 3—9.6 lb. Total with 12-in. pipe at Section 3—5.4 lb.			Total with 10-in. pipe at Section 3—8.8 lb. Total with 12-in. pipe at Section 3—6.4 lb.		
Aver. using 10-in. pipe in Sections 2 and 3 = 10.8 lb. Aver. using 12-in. pipe in Sections 2 and 3 = 6.4 lb.						Aver. using 10-in. pipe in Sections 2 and 3 = 9.8 lb. Aver. using 12-in. pipe in Sections 2 and 3 = 5.4 lb.						Average as above.		

section will depend upon the location of the compressors; i.e., whether all the compressors are located at one shaft or whether the installation is split between the two shafts. If the entire compressor installation is installed at one of the shafts, then section No. 1 will have to carry 7,500 c.f.m. of free air and should be 12 in. in diameter. If the installation is split and a portion of the capacity placed on the other shaft, the line may be made smaller. However, since it must be large enough to carry more than a normal amount of air in case one of the units is shut down, it should not be smaller than 8 in. in diameter.

After the compressors had been chosen, it was decided to install them all at No. 2 shaft and a 12-in. pipe was therefore needed in section No. 1.

TABLE 3—PRESSURE AIR COMPRESSORS MUST MAINTAIN

	POINT A		POINT B		POINT C	
	USING 10-IN. PIPE-SEC. 3 LB.	USING 12-IN. PIPE-SEC. 3 LB.	USING 10-IN. PIPE-SEC. 2&3 LB.	USING 12-IN. PIPE-SEC. 2&3 LB.	USING 10-IN. PIPE-SEC. 2&3 LB.	USING 12-IN. PIPE-SEC. 2&3 LB.
Distribution point press....	85	85	85	85	85	85
Friction Loss..............	+8.8	+6.4	+10.8	+6.4	+9.8	+5.4
Pressure Gain.............	−10.3	−10.3	−14	−14	−14	−14
Pressure at compressor.....	83.5	81.1	81.8	77.4	80.8	76.4

Average pressure at compressor using 10-in. pipe at sections 2 & 3 = 82.0 lb.
Average pressure at compressor using 12-in. pipe at sections 2 & 3 = 78.3 lb.

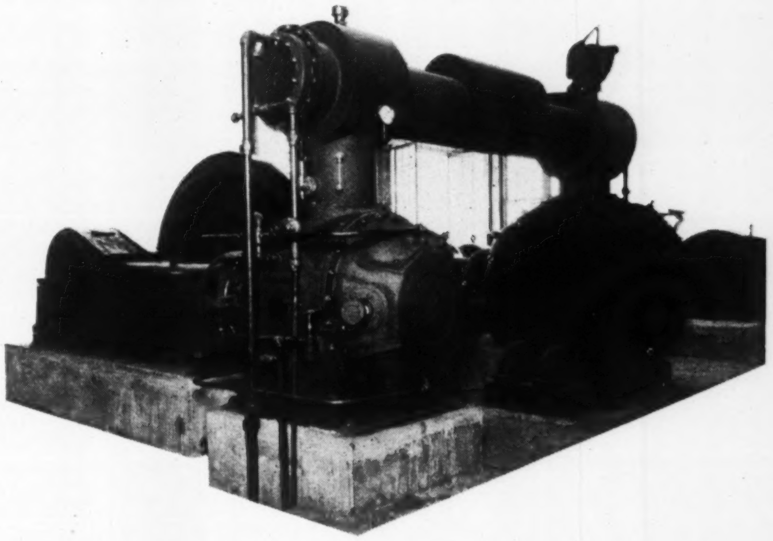

FIG. 5—A SYNCHRONOUS MOTOR DRIVEN AIR COMPRESSOR OF THE TYPE USED BY THE MINE MENTIONED IN THIS ARTICLE

Comfort Zone in the Southwest

By a Reader in Texas

Air conditioning in the Southwest as in the North, East and other parts of the country is as yet a new science; and many of us are "practicing" as our brother scientists, the doctors, put it. We, in the Southwest, are dependent to a great extent on the North and East for our new ideas and thoughts due to the fact that the centers of research and manufacturing are there.

In air conditioning work for summer conditions, we are called on to figure systems that will maintain conditions within the comfort zone as specified by the *American Society of Heating and Ventilating Engineers.* It has been found that the comfort zone in this territory is slightly different from that in the East.

The lower limit has not been checked, but the upper limit has and the results, though quite variable, show that the line defining the upper limit should be a curve beginning at 69 degrees saturation and passing through 75 deg. dry bulb and 60 per cent relative humidity; 80 deg.—40 per cent; 85 deg. dry bulb—30 per cent.

Why these conditions are comfortable is best explained by calling to the reader's attention the climatic conditions to which the Southwest is subjected. Our hot weather extends over a period of seven months, the balance of the year is cool, but never as cold as the North and East. Even in our winter, we frequently have days during which the temperature rises to 85 deg. dry bulb. Naturally this preponderance of warm weather has a great influence on the thickness of blood and the sensitiveness of the skin.

On an average day in the summer with a dry bulb temperature of 95 deg. and 40 per cent relative humidity, if a person enters a room in which the dry bulb is 80 deg. and the humidity is 25 per cent, a distinct chilliness is felt. On leaving after an hour or so, the change to the outside conditions is depressing. Women have been known to faint and in most cases the effect on everyone is uncomfortable. During these hot months conditioned air with a 10 to 15-degree drop in dry bulb temperature and a slightly higher percentage of humidity is much more comfortable in this territory.

Editorial Note: Comment on the above communication from a Texas reader is solicited.

By
Ludwig Wahl,[1]
Paul Reschke,[2]
and F. E. Giesecke[3]

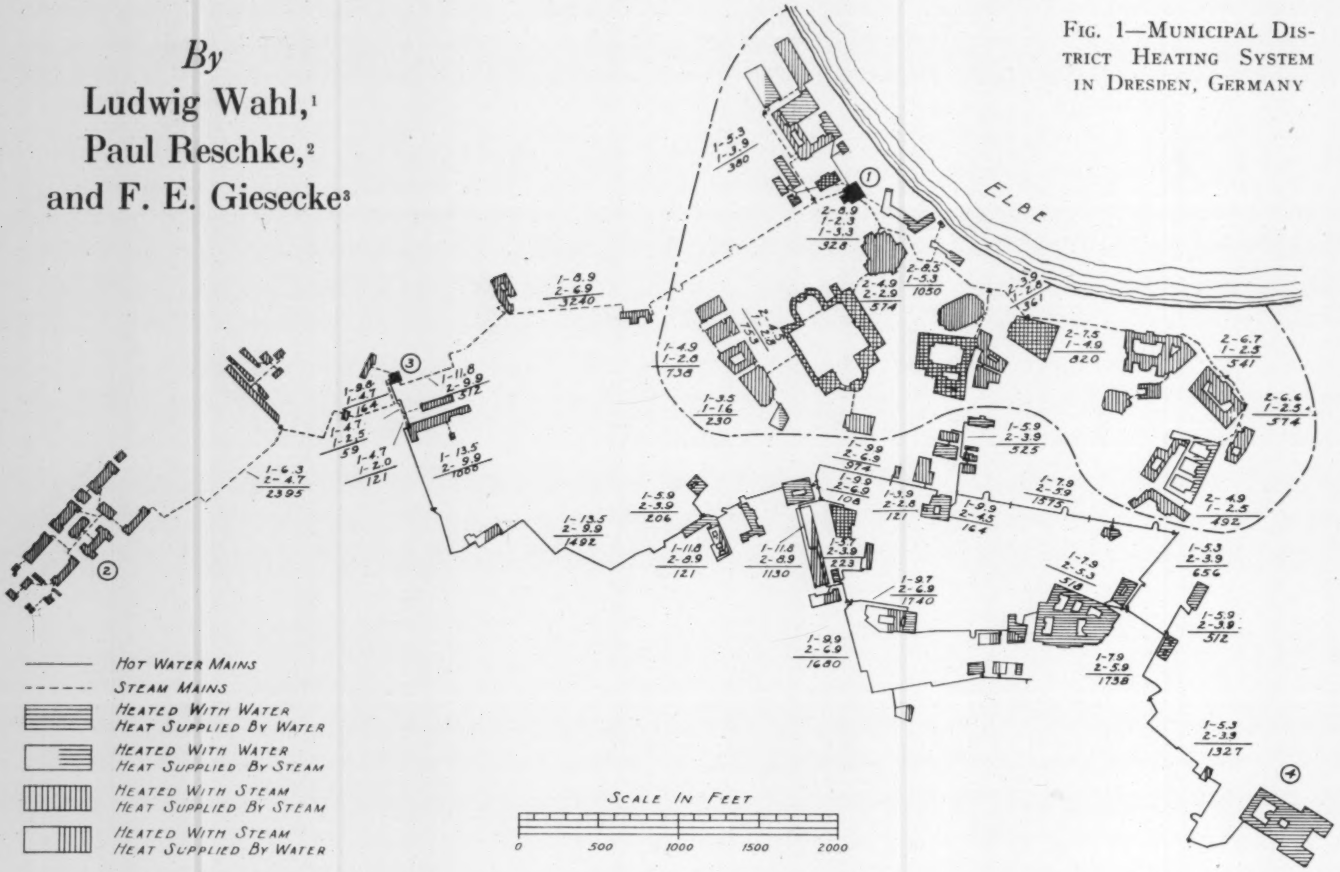
ELBE
①
②
③
④
1-5.3 1-3.9 380
2-8.9 1-2.3 1-3.3 328
2-8.5 1-5.3 1050
2-4.9 2-2.9 574
2-3.9 1-2.8 361
2-4.3 1-2.8 753
1-4.9 1-2.8 738
1-3.5 1-1.6 230
2-7.5 1-4.9 820
2-6.7 1-2.5 541
2-6.6 1-2.5 574
1-5.9 2-3.9 525
1-9.9 2-6.9 974
1-9.9 2-6.9 108
1-3.9 2-2.8 121
1-9.9 2-4.5 164
1-7.9 2-5.9 1575
2-4.9 1-2.5 492
1-8.9 2-6.9 3240
1-11.8 2-9.9 572
1-9.8 1-4.7 164
1-4.7 1-2.5 59
1-4.7 1-2.0 121
1-13.5 2-9.9 1000
1-6.3 2-4.7 2395
1-13.5 2-9.9 1492
1-5.9 2-3.9 206
1-11.8 2-8.9 121
1-11.8 2-8.9 1130
1-5.7 2-3.9 223
1-9.7 2-6.9 1740
1-9.9 2-6.9 1680
1-7.9 2-5.3 518
1-5.3 2-3.9 656
1-5.9 2-3.9 512
1-7.9 2-5.9 1738
1-5.3 2-3.9 1327
Hot Water Mains
Steam Mains
Heated With Water Heat Supplied By Water
Heated With Water Heat Supplied By Steam
Heated With Steam Heat Supplied By Steam
Heated With Steam Heat Supplied By Water
Scale In Feet
0
500
1000
1500
2000

Fig. 1—Municipal District Heating System in Dresden, Germany

District Heating System Supplements Municipal Power Plant

Improves Over-all Efficiency

> In this hot water district heating system, the water is circulated at so high a temperature that those buildings which require steam for heating, cooking, sterilization, etc., can be supplied with steam from heat exchangers arranged so that the hot water circulates through the coils and steam is generated in the space surrounding the coils. The system has several other outstanding features. ▼ ▼ ▼

The first European district heating and lighting system of considerable magnitude was installed in the City of Dresden in 1900-1902 by the State of Saxony according to plans prepared by Professor Pfutzner. This system is shown in the upper portion of Fig. 1 and is inclosed by a dashed and dotted line. Its power house is shown as No. 1. The original purpose of this plant was to provide several state buildings with light and heat; later, the service was extended to include a number of private buildings.

During the early period of the operation of this system, steam was generated at a pressure of about 118 lb. and used in part to operate steam engines driving electric generators and in part to heat buildings by means of a central system in which the steam was distributed by mains having a maximum length of 4,000 ft.

In 1911 the district steam heating system was supplemented by a district hot water heating system, in order to increase the thermal efficiency of the power plant. The water was circulated by centrifugal pumps. It was passed through the condensers in the power house where it served as cooling water and where its temperature was brought to about 158 F. From the condensers the water passed through heat exchangers in which the temperature was raised to about 200 F.

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[2] Dr. Ing., Engineer, Dresden.
[3] Director, Texas Engineering Experiment Station, A. & M. College of Texas, College Station.

In addition to the state power plant, Dresden had a municipal plant which was built about 1900 and which supplied electric current for the street railway system, for the illumination of the streets, and for other uses. The equipment of this plant included seventeen boilers with a combined heating surface of 24,250 sq. ft. and two turbo-generators with a combined capacity of 16,250 kva. at 2,000 volts.

Demand for Current Increases

During recent years the demand for current in Dresden has increased rapidly. In 1914 it was 40,400,000 kw-hr.; in 1924, it had grown to 69,700,000, and in 1925 to 98,200,000. A considerable portion of the current was being supplied by other than the municipal plant; this arrangement was not satisfactory and so, in 1925, preliminary steps were taken looking to the enlargement of the municipal plant.

Heating System Improves Over-all Efficiency

At the suggestion of one of the authors,[4] it was decided to supplement the new power plant by the addition of an extensive district heating system so as to secure a greater over-all efficiency of the installation. Satisfactory experience with such a combination had already been gained in the heating of the municipal hospital, shown in Fig. 1 as No. 2. This institution had a heat demand of 12,735,000 Btu per hr., which was supplied from the municipal power house, shown in Fig. 1 as No. 3, with steam at 44 lb.

The construction of the new boiler house was begun in June, 1927. Five new boilers were installed, each having a heating surface of 3280 sq. ft., and designed to supply steam at 544 lb., superheated to 770 F.

Hot Water System Selected

In connection with the selection of the most suitable steam pressure stages for the power plant, a thorough study was made of the relative advantages of steam and water as heat conveyors for the proposed district heating system.

A hot water system was selected and its construction was begun soon after that of the new power house. The full lines in Fig. 1 show the mains. The dotted lines represent steam mains. One extends from the new power house (No. 3) to the hospital group (No. 2) and the other to the former state power house (No. 1) from where the heat is distributed partly by steam mains and partly by water mains as shown in Fig. 1.

The total length of the water mains leading out of the new power house is about 16,000 ft.; the most remote building served is the German hygiene museum (No. 4).

The figures along the mains show the sizes and the lengths of the respective sections. For example, 1 × 13.8, 2 × 9.8 and 1,490 means that this particular section of the main consists of one 13.8-in. and two 9.8-in. pipes and that each pipe has a length of 1,490 ft.

Features of the System

Among the outstanding features of this hot water district heating system are:

The distributing mains are composed of three pipes valved so that any two may serve as flow and return mains in such a manner that repairs or additions may be made at any time without interrupting the regular service.

The water is circulated at so high a temperature that those buildings which need steam for heating, or cooking, or sterilization, etc., can be supplied with steam from heat exchangers arranged so that the hot water circulates through the coils and the steam is generated in the space surrounding the coils.

The quantity of heat supplied a customer is metered by instruments which measure the quantity of water flowing through the coils of the heat exchanger and also the drop in temperature of the water as it flows through the heat exchanger; these two quantities are multiplied and integrated mechanically so that the meter reading is in Btu-hours.

Two Flow Lines; One Return Line

Fig. 2 shows four principal types of concrete conduits for the three pipe lines composing the distributing mains.

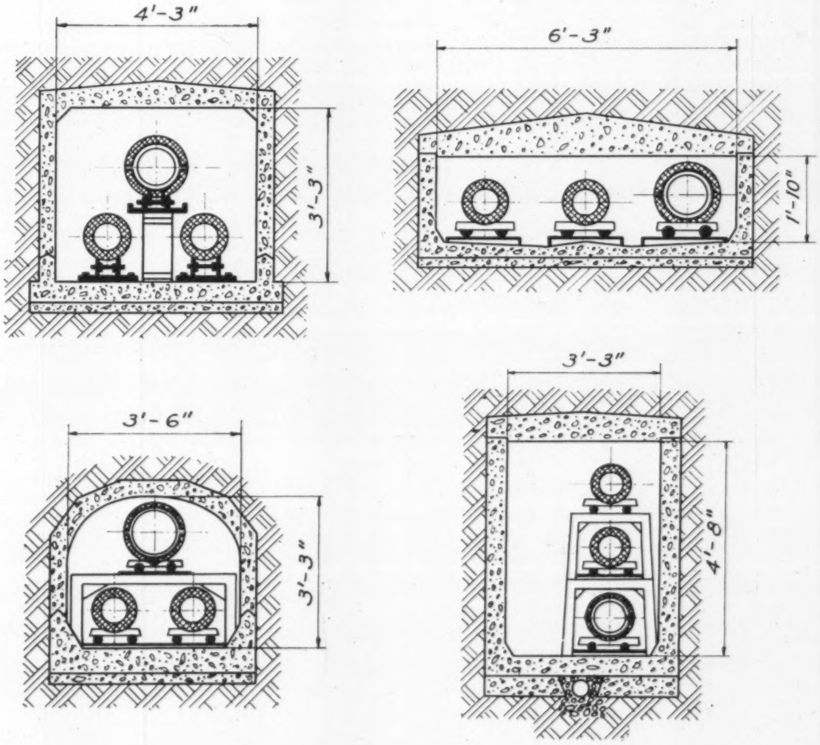

Fig. 2 — Sections of Concrete Pipe Conduits

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[4] Dr. Ing. Wahl.

As ordinarily operated, the large pipe is the return line and the two smaller are the flow lines. The cross-sectional area of the large pipe is practically equal to the cross-sectional areas of the two smaller pipes.

One of the flow lines is intended to convey heat to those buildings which are equipped with hot water heating systems. This line will convey 257-F water except in extremely cold weather when the temperature of the water will be higher. The other flow line is intended to convey heat to those buildings which are equipped with low pressure steam heating systems; this line will always convey water having a temperature higher than 257 F.

At the present time, the heating system is being operated at a fractional part of its capacity and in order to reduce its heat losses as much as possible, only one of the two flow lines is being used. It conveys water having a temperature of about 284 F.

In the return line the water has a temperature of about 167 F. The temperature drop through the system is about 117 F and one cubic foot of water conveys about 7,000 Btu.

80-Lb. Static Head

The hot water is circulated under a static head of about 80 lb. per sq. in. This pressure is maintained either by open expansion tanks located in the tower of the city hall at an elevation of about 200 ft., or by a closed expansion tank located in the power house, in which an air pressure of approximately 75 lb. per sq. in. is maintained. The open expansion tanks have a combined capacity of 21,000 gallons; they are connected to the return main about 8,000 ft. from the power house but the pressure is so great that the circulating pumps function properly.

The water is circulated by four centrifugal pumps with a combined capacity of about 3,000 g.p.m. When the system is being operated at its proposed maximum capacity the velocity of the water in the mains will be 9.84 ft. per sec.

The friction head in the system is approximately 37 lb. per sq. in.; with future additions to the system, this will probably be increased to 80 lb. per sq. in.

Heating the Water

The water is preheated by four heat exchangers having a combined capacity of 90,000,000 Btu per hr. and superheated by two heat exchangers having a combined capacity of 20,000,000 Btu per hr. About 85 per cent of the steam used for heating the water is at a pressure of 26 lb. and the balance is at a pressure of 220 lb.

The Underground Piping

While most pipes are placed in concrete ducts as shown in Fig. 2, some of which are large enough so that a person can crawl through them, recent installations have been made without ducts, the pipes being placed directly into the ground. In these cases, the pipes expand in an axial direction within the insulation which is protected against the earth either by lead-coated sheet metal thoroughly asphalted inside and outside and in a jute casing, or by a reinforced concrete casing.

Where concrete ducts are used they are sloped in the same manner as the pipe lines. At the low points of the ducts provision is made for the removal of water either by direct connection to sewer or by means of sumps provided with automatic pumps. At the low points of the pipe lines valves are placed so that the pipes can be drained. At the high points of the pipe lines air chambers of ample size are provided and connected to vent lines. Manholes are placed at all points where the direction of the mains changes and provisions are made so that the sections between manholes can be illuminated by means of electric lights.

Pipe lines placed directly in the ground are provided with expansion loops and with observation manholes at all points where the direction changes.

Expansion of the lines is taken care of, so far as possible, by proper changes in direction. Where this is not possible, expansion loops are used, built either of plain or corrugated pipe bends, or metal hose compensators are inserted. By these means expansions as large as 18 inches are provided for.

The material selected for pipe insulation is spun glass or spun slag. It was applied to the pipes and then covered with sheet metal, cement, or a special preparation coated with bitumen.

Connections to Building with a Hot Water System

Throughout the entire system, the mains consist of two flow lines and one return line, all three lines extending to the distribution panels in the buildings to be heated. At these distribution panels a group of five valves is provided as shown in Figs. 3 and 4, arranged so that, at any time, any two of the three mains may be used to circulate hot water through the heat exchangers of the building.

Fig. 3 shows the connection of the mains to a building provided with a hot water heating system. The five valves are set so that the heat exchanger receives its water from flow line II, conveying water at a temperature of 257 F, or more, and returns it through the return line.

The hot water from the power plant does not circulate through the heating system of any building. It is used only to convey heat from the power house to the several buildings.

This induced transfer of heat is necessary because the private heating system may not be of sufficient strength to safely withstand the high pressure maintained in the district heating system, and it is of great advantage because leaks in the private heating system cannot drain water from the district heating system.

Connections to Building with a Steam System

Fig. 4 shows the connection of the mains to a building provided with a steam heating system. In this case the five valves are set so that the heat exchanger receives its water from flow line I, conveying water at a temperature of 284 F, or more, and returns it through the return line. Fig. 4 also shows the auxiliary apparatus as does Fig. 3.

The heat exchanger, in this case, has the hot water from the district mains flowing through its coils; the space around the coils is only partly filled with water, so as to leave an exposed water surface from which the

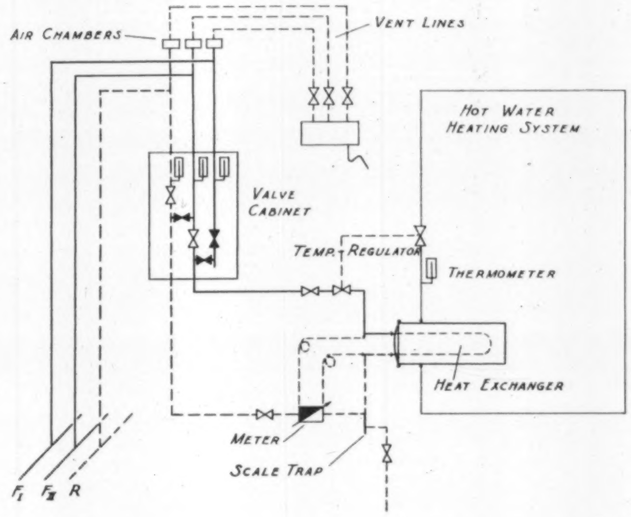

Air Chambers
Vent Lines
Hot Water Heating System
Valve Cabinet
Temp. Regulator
Thermometer
Heat Exchanger
Meter
Scale Trap
F_I
F_{II}
R

Fig. 3—Arrangement of Valves for Hot Water System to Which Heat Is Supplied by Means of Hot Water

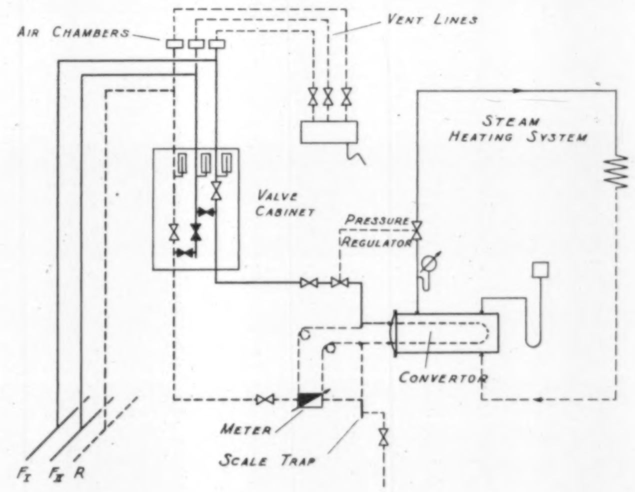

Air Chambers
Vent Lines
Steam Heating System
Valve Cabinet
Pressure Regulator
Convertor
Meter
Scale Trap
F_I
F_{II}
R

Fig. 4—Arrangement of Valves for Steam Heating System to Which Heat Is Supplied by Means of Hot Water

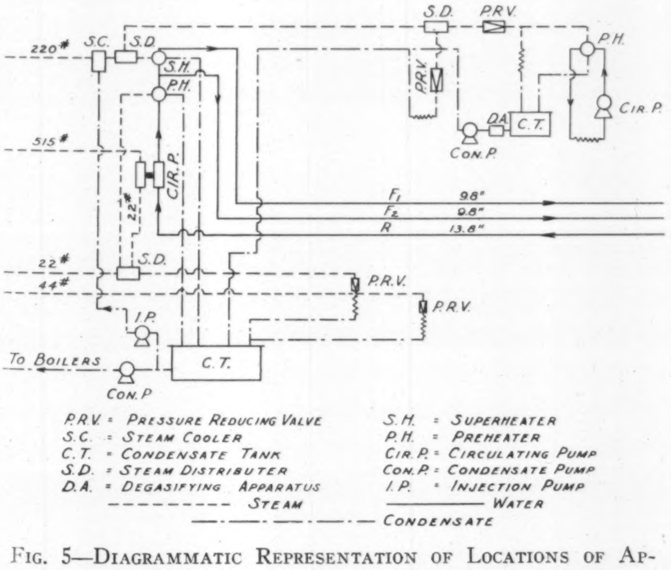

S.D.
P.R.V.
220#
S.C.
S.D.
S.H.
P.H.
P.H.
P.R.V.
Cir. P.
D.A.
C.T.
Con. P.
515#
CIR. P.
22#
F_1 9.8"
F_2 9.8"
R 13.8"
22#
S.D.
44#
P.R.V.
P.R.V.
I.P.
To Boilers
C.T.
Con. P
P.R.V. = Pressure Reducing Valve
S.C. = Steam Cooler
C.T. = Condensate Tank
S.D. = Steam Distributer
D.A. = Degasifying Apparatus
S.H. = Superheater
P.H. = Preheater
Cir. P. = Circulating Pump
Con. P. = Condensate Pump
I.P. = Injection Pump
Steam
Water
Condensate

Fig. 5—Diagrammatic Representation of Locations of Apparatus in Distribution Chamber of Power House

steam for the heating system is liberated; the heat exchanger is the steam boiler for the building heating system.

The pressure of the steam in the private steam heating system, as well as the temperature of the water in the private water heating system, are regulated by hand-operated valves or by thermostats.

The water which returns from the heating system to the power house is forced by steam-driven centrifugal pumps through heat exchangers in which the water is preheated to a temperature of about 257 F by 26-lb. bleeder steam from the steam turbines of the generating sets. The 257-F water can be delivered directly into the flow mains of the heating system, or it can be passed through superheaters, in which the temperature is raised by 220-lb. steam and delivered from there into the flow mains.

Connecting New Customers

Sets of valves like those shown in Figs. 3 and 4 are placed in every building served by the district system and also at other important points. By means of these valves, it is possible to discontinue the use of any portion of any one of the three mains at any time in order to make repairs or in order to connect new customers without interfering with the service of any portion of the system. This arrangement has been found very satisfactory as new customers are being added continually.

Arrangement of Piping in Distribution Chamber

The arrangement of the piping, valves, and distributors in the distribution chamber of power house No. 2 and the connections of the station to the three district heating systems shown in Fig. 1 are illustrated in Fig. 5, and described briefly as follows:

Six lines enter the distribution chamber, coming either from the boiler house or from the engine room; two of the lines are 3.94 in. and convey 515-lb. steam, two are 11.81 in. and convey 220-lb. steam, and two are 23.62 in. and convey 22-lb. steam. The corresponding lines are united in double distributors constructed so that the two halves of each distributor are connected by gate valves. The 515-lb. steam is used only in the turbines of the circulating pumps of the hot water heating system. It is extracted from these turbines at a pressure of 22-lb. and passed into the distributors. The steam is transmitted from the distributors to the several stations where it is to be used.

All important valves, machines, meters, and control apparatus are located on one floor so that they can be conveniently attended by one operator.

Condensate

Condensate from the entire system is collected and returned to the boilers; a small portion is used in steam coolers which are built into the 220-lb. steam distributors and in which the steam—which has been superheated to about 335 F—is cooled to about 250 F.

The larger part of the condensate of the 220-lb. steam heating system from power house No. 1 is still degasified. The condensate from the heaters for the hot water heating system is not degasified, as it has no opportunity to absorb air. The condensate tanks are covered with a blanket of steam to prevent absorption of air; a possible small air content is removed in the preheating of the feed water.

Heat Demand Increases Rapidly

The heat demand of the hot water heating system was 10,000,000 Btu per hr. at the beginning of the heating season of 1928; it grew to 27,500,000 during the season 1928-29, and to 67,200,000 by February, 1930. Of this quantity, 23,200,000 were for buildings equipped with steam heating systems and 44,000,000 for buildings equipped with hot water heating systems, or for domestic hot water heating. The principal buildings supplied with heat are the city hall, schools, post offices, German hygiene museum, stores, banks, industrial buildings, and places for amusement.

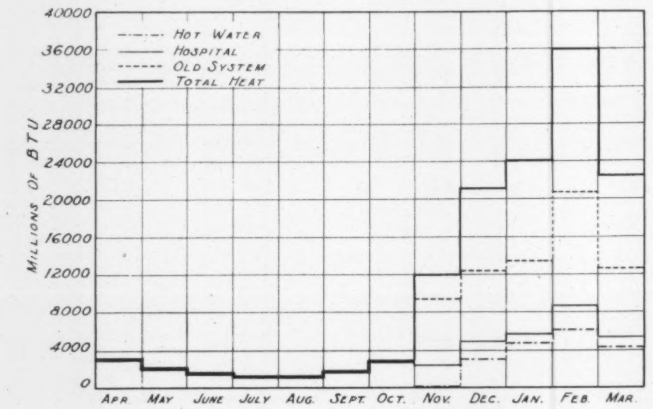


Fig. 6 — Quantities of Heat Sold in 1928-29

State Power Plant Abandoned

After the completion of the new power plant the city of Dresden acquired the state power plant and discontinued its operation since it was no longer able to meet the demands upon it. To provide heat for the districts formerly served by it, pipe mains were laid from the new power house to the old—a distance of about 3,500 ft.—and supplied with steam from the 220-lb. distributors. They consist of one 9.8-in. flow line, one 6.9-in. flow line, and one 6.9-in. return line. In case of necessity, the return line can be used as a flow line. These pipe sizes were determined so that 60 tons per hour of steam can be conveyed when the available drop in pressure (from 220 to 118 lb.) is completely absorbed.

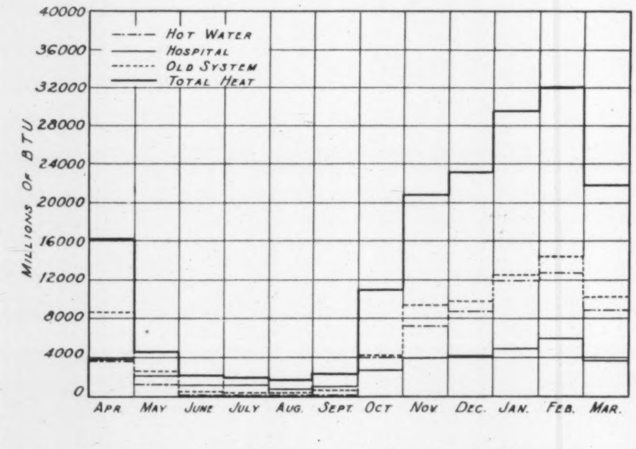


Fig. 7 — Quantities of Heat Sold in 1929-30

The total heat demand of power house No. 1 is 114,000,000 Btu per hr. Since the hot water lines from power house No. 1 were originally installed to increase the thermal efficiency of that power plant, and since the operation of that power plant has been discontinued, all new installation will be for steam and eventually only steam heating lines will radiate from power house No. 1.

Total Heat Delivered

Figs. 6 and 7 show the total heat delivered during the several months of the cold winter of 1928-29 and also during that of the moderate winter of 1929-30; these diagrams show that the heat delivered during February, 1929 was about 36,000,000,000 Btu and during February, 1930, about 32,000,000,000 Btu.

The diagrams show also how the total heat delivered was distributed by the three main subdivisions of the heating system.

Variation in Production of Steam and Electricity

Figs. 8 and 9 show the variation in the production of steam and of electric current during two typical days, one a summer day when the demand for heat was small, and the other a moderate winter day.

In the summer the heating system is used only in hospitals, hotels, theaters, and similar buildings where hot water is needed for domestic or industrial purposes. During winter days, the peak demand is at about 8 p. m., and is produced practically in equal parts by the high pressure steam lines and by the hot water lines. The total heat delivered during the summer days is also

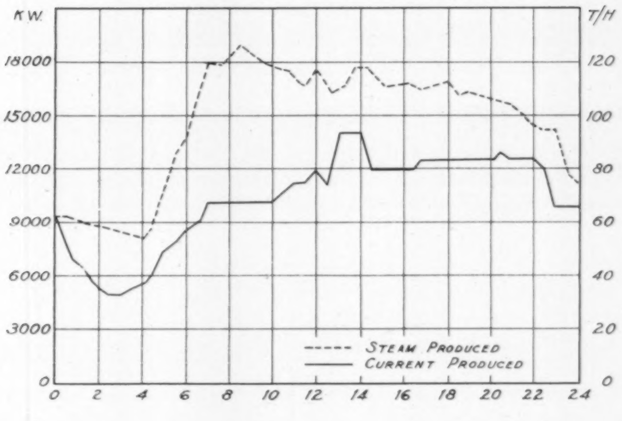


FIG. 8 — QUANTITIES OF STEAM AND CURRENT PRODUCED JANUARY 31, 1930

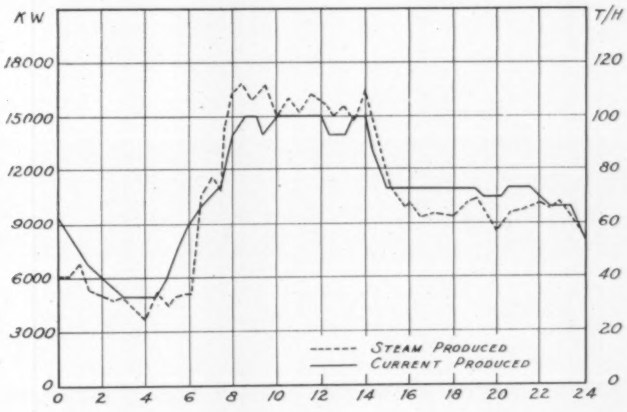


FIG. 9 — QUANTITIES OF STEAM AND CURRENT PRODUCED June 7, 1930

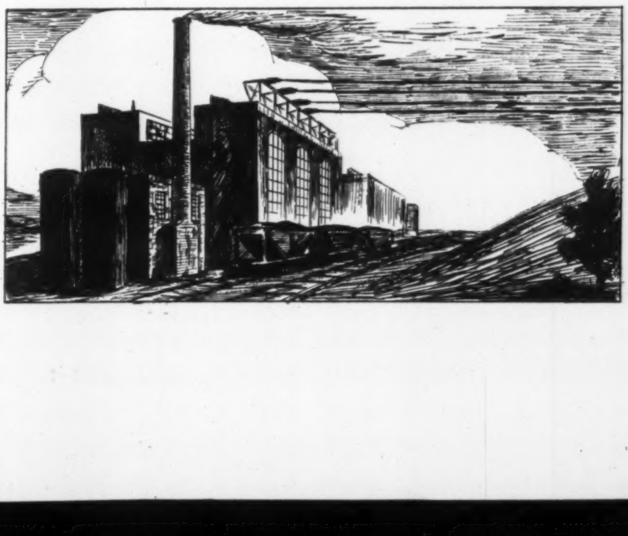
practically equal for the two systems, although the maximum demand of the high pressure steam lines is almost twice as great as that of the hot water lines. This shows that the factor of use of the high pressure steam lines is much lower than that of the hot water lines; this is brought about by the difference in the users of the two types of heating. In the first case they are museums, theaters, and churches, and, in the second case they are primarily business, industrial, and administration buildings.

Hospitals Good Customers

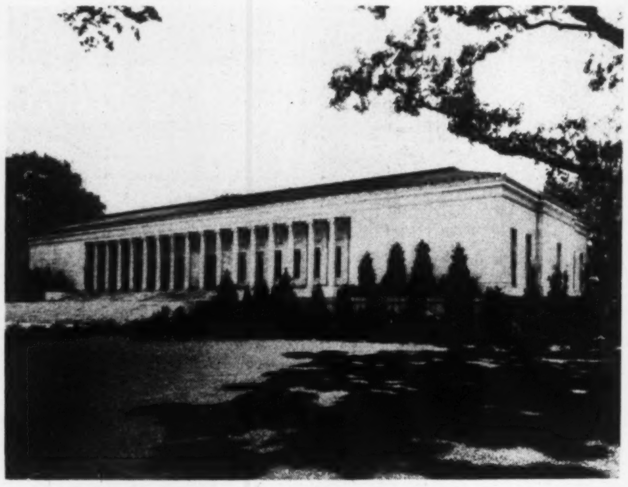
A very satisfactory customer, from an economic point of view, is the hospital group; it begins taking heat at 4:30 a. m. and uses it at a practically constant rate from 6 a. m. to 6 p. m. The unusual economic quality of customers who use considerable quantities of heat during the summer is shown by the fact that the hospital group used 17,650 tons of steam during the year 1928-29, whereas the entire high pressure steam system, which had a demand six times as large, used only 38,000 tons, or only about twice as much.

If the energy supplied by the heating systems is expressed in terms of hours of demand, there are, for the hospital, 2100 hours, and for the high pressure steam lines, only 760 hours per year. In the case of the hot water heating system, the average number of demand hours supplied is about 1100 per year. In considering these figures it is necessary to bear in mind that the winter of 1928-29 was the coldest in about 100 years. During normal winters the number of hours supplied will be reduced at least ten per cent for those customers who use the heat only for heating buildings.

Editor's Note: In the design and execution of the plans for the district plant Dr. Wahl had the able assistance of Dr. Ing. Paul Reschke, as well as that of the engineering staff of the municipal power plant. The installations were made by Dresden firms, principally Wilh. Vocke, Rietschel & Henneberg, and Jeglinsky and Tichelmann.



By Samuel R. Lewis*



THE TOLEDO ART MUSEUM BEFORE THE RECENT ADDITIONS WERE MADE

Heating and Humidifying Art Museums

a Description of the Features of the Toledo Museum of Art

ART MUSEUMS are notorious for the difficulties encountered when it comes to serving their galleries with heat. The ceilings usually are of glass to admit sunlight, or to permit artificial lighting by reflectors placed above them.

These arrangements militate strongly against the running of any ducts through the attic spaces above the gallery ceilings and below the skylights. The attics, in any event, usually must be kept clear of ducts so as to permit walkways and passages in order that the lamps may be serviced and the glass ceilings cleaned. All side walls in the galleries above a line about 30 in. above the floor must be reserved for exhibits.

The floors usually are of expensive and beautiful material and floor registers are undesirable. Thus, very few areas suitable for radiators or registers are left.

In some galleries there are center seats, and it is possible to use the spaces under these—and behind their backs—for radiators and for air supply and exhaust grilles. However, if the room under the gallery is of value, and usually it is, the pipes and ducts there, necessary to serve a gallery equipped with center radiators and grilles, are exceedingly obnoxious.

Also, while center seats may do very well for a picture gallery, they are usually not suitable if the gallery is used for exhibits of ceramics, jewels, etc., which are placed in cases. In any gallery it is usually desirable—sooner or later—to change the nature of the occupancy. If it is desired to install or to remove the center seats, this should be possible without any complications arising due to the general service pipes and ducts of the building.

Pre-warmed Air Introduced at Doors

A solution well worked out in one large museum of art is to introduce pre-warmed air at low velocity into each gallery through the soffits of the door openings, venting the air out at the side walls under the lower limit of the hanging line mentioned above, about 30 in. from the floor. (Since this line must be mentioned often, it is going to be called the chair-rail for convenience, hereinafter, because it is more nearly like a chair-rail than anything else.)

This location of air supply inlets at the doorways is fairly satisfactory, but the air distribution within the gallery when it is employed is not usually perfect or symmetrical, and the air can be felt coming as one passes through the doorway.

Temperature control with this scheme is rather diffi-

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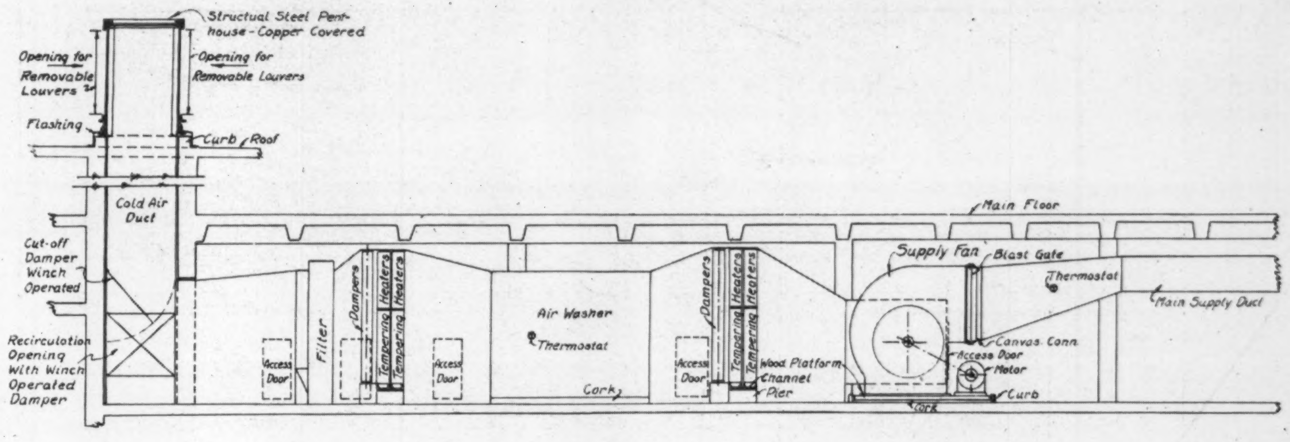
*Consulting engineer, Chicago, Ill.


Structual Steel Pent-house-Copper Covered
Opening for Removable Louvers
Opening for Removable Louvers
Flashing
Curb
Roof
Cold Air Duct
Cut-off Damper Winch Operated
Recirculation Opening With Winch Operated Damper
Access Door
Filter
Dampers
Tempering Heaters
Tempering Heaters
Access Door
Air Washer
Thermostat
Cork
Main Floor
Access Door
Dampers
Tempering Heaters
Tempering Heaters
Wood Platform
Channel
Pier
Supply Fan
Blast Gate
Thermostat
Main Supply Duct
Canvas Conn.
Access Door
Motor
Curb
Cork

VERTICAL SECTION THROUGH AIR INTAKE SHAFT AND AIR-PREPARING EQUIPMENT

Cold air from outside or return air from the attic passes down the shaft and encounters the automatic oil-immersed dust filters, the tempering heaters, the air washer, the retempering heaters and the supply fan. The supply fan delivers the air out to the reheaters for the individual rooms which are always under, or in, the rooms.

cult if two adjacent galleries happen to have different rates of heat loss, especially if the door between them is open.

Inlets on Walls

In some galleries appearances are disregarded to some extent and the inlet openings are placed in the conventional location, on the inside walls opposite the windows or coolest walls, and about eight feet above the floor. When this is done there can be no pictures in that location and the velour with which the walls usually are hung quickly becomes dusty.

Long Radiators in Recesses

The original Toledo Museum of Art, which embraced about one-half of the center building of the 1931 project, or very roughly, one-sixth of the total building, had the center seat arrangement in its galleries. There was no ventilation provided for the galleries, although they had no windows. The hemicycle, or auditorium, was equipped with a mechanical ventilating system.

In constructing the 1923 portion, the first addition to the original building, a basic requirement by the owner was that all of the galleries be ventilated mechanically and the radiators and center seats be removed from many of the galleries.

The solution developed in 1923 was to install a very low, thin and lengthy radiator in a recess almost as long as the width of the room at the coldest end of each gallery, behind a grille and below the chair-rail. The air supply to the gallery from the fan system was delivered around the radiator and through this grille. At the opposite end of the room the outlet grille, similarly located and looking just like the inlet grille, was installed. There were a number of objections to this scheme which developed as experience was gained in operation, mainly overheating of the wall and consequent damage to exhibits.

A blast of warm air, not very hot, it is true, but very potent in ability to absorb moisture from the exhibits within its range, came through the grille. The wall above the radiator and the entire neighborhood were objectionably warm.

Experience with the 1923 galleries where the warm air entered at one end near the floor and escaped at the other end near the floor was satisfactory during the winter, but was unsatisfactory during hot weather. When the entering air was cooler than the air in the room it passed from inlet to outlet, cooling the knees of the visitors and not disturbing the hot moist air, which, (being lighter than the cool air) remained above the chair-rail zone and became more and more torrid as the sun developed its effect on the glass skylights and ceilings. It was proved by experiment that low-down inlets were unobjectionable in hot weather, but that low-down outlets in connection with low-down inlets were ineffective in hot weather.

When the low-down outlets were closed and a sheet of ceiling glass was removed in hot weather however, the cool, relatively heavy air filled the lower part of the gallery and forced the hot air upward above it until the hot air could escape through the skylight into the attic.

Extensive Studies Made with Models

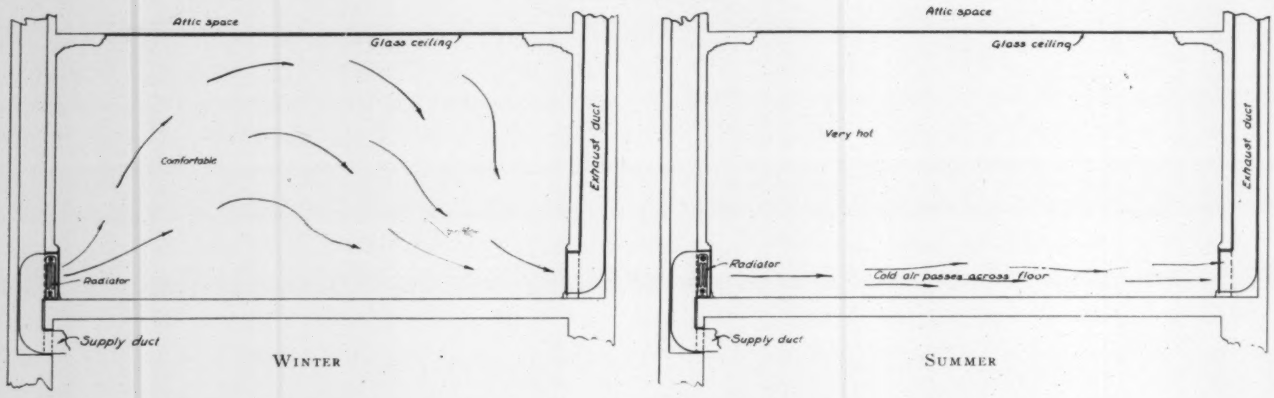
Following this experience, when the 1931 additions were planned, considerable experimenting and research was done, and many life-size working models were made in an endeavor to improve the methods of air introduction and removal and to develop a better way to heat and to ventilate the museum.

Change in Humidity Deteriorates Exhibits

From all that could be learned, based on our own experience and that of the Bureau of Standards, the *exact* moisture content and the *exact* temperature within reasonable limits, are of no particular importance *so far as the preservation of the books and antiquities are concerned.* The condition which ages them unduly and which causes cracking and splitting and deterioration is a rapid *change* in the moisture content. They suffer far more each spring and each fall when artificial heating stops and when it begins than they do during the time after the heated condition or the unheated condition has become established.

Air Is Now Humidified

It was decided, therefore, to install means of introducing artificial moisture during these critical periods so


Attic space
Glass ceiling
Comfortable
Exhaust duct
Radiator
Supply duct
Attic space
Glass ceiling
Very hot
Exhaust duct
Radiator
Cold air passes across floor
Supply duct

WINTER SUMMER

VERTICAL SECTIONS THROUGH A GALLERY SHOWING ABANDONED 1923 HEATING AND VENTILATING ARRANGEMENTS
The circulation of air was satisfactory in winter but in summer the incoming cool air would short-circuit.

as at least to lengthen the time of transition, and so as to provide year-round artificial moisture if desired, to simulate the higher summer relative humidity. Provisions were made to facilitate the future installation of dehumidification apparatus.

It was decided that there should be no radiators nor any piping carrying water or steam inside any gallery, in order to run no chance of destroying priceless exhibits by an accidental leak.

Warm Air Delivered Close to Floor

It was found by experiment with the working models that a rather high velocity jet of warm air could be delivered close to the floor in a horizontal direction through the baseboard without prejudice to the visitors and with the least prejudice to the exhibits. A 300-f.p.m. jet of warm air would clear all pictures hanging on the wall above it, yet would not be objectionably hot for show cases placed four or five feet from the jet and above its direct influence.

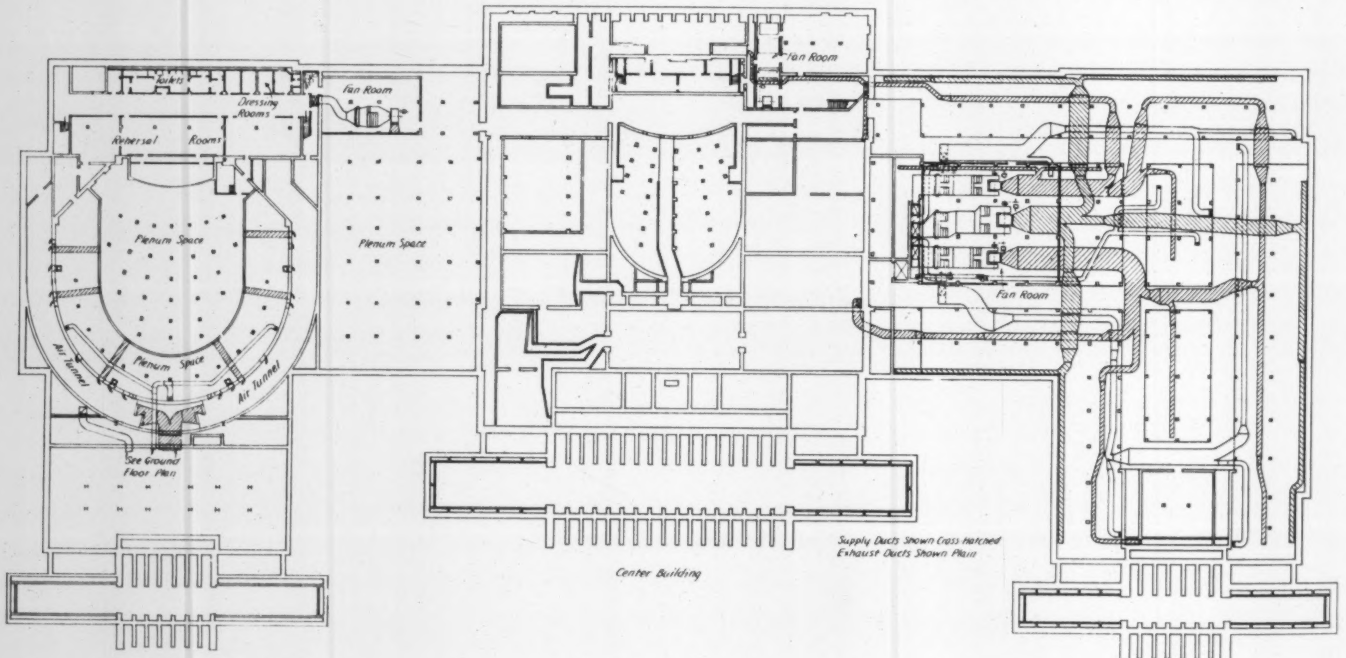
Pressed Steel Wainscot Developed

A standard pressed steel wainscot was developed, embracing in itself the chair-rail and the baseboard and the inlet grille; this steel base is used throughout all galleries, whether or not in any given gallery all of the wainscot length is used for the air inlet. Usually sufficient grille area is obtainable by using the wainscot on two sides of each gallery.

Arrangement of the New System

Art museums usually have furred-out wooden walls under the cloth surfaces, to facilitate hanging pictures of various sizes safely. These furred-out spaces in the Toledo museum are used to run horizontal distributing ducts of metal under them to supply the wainscot slot-inlets. In general, each 16-ft. bay of the building throughout the gallery story has a separate supply duct and is capable of receiving air at a different temperature from that delivered to adjacent bays.

The warmed and conditioned air, after being delivered


Toilets
Dressing Rooms
Rehersal Rooms
Fan Room
Plenum Space
Plenum Space
Plenum Space
Air Tunnel
Air Tunnel
See Ground Floor Plan
Fan Room
Fan Room
Supply Ducts Shown Cross Hatched
Exhaust Ducts Shown Plain
Center Building
East Wing
West Wing

BASEMENT PLAN

The original building includes approximately half the front of the center building. The 1923 addition includes the back part of the center building. The 1931 additions include the west wing, to the right, the east connecting building and the music hall to the left. The basement in general is used for storage and for pipes and ducts. It is entirely below ground level.

into the gallery at the floor—in most cases, throughout the length of the rooms on both sides—rises and escapes through the ceiling to the attic.

There are no visible vent grilles in the glass ceiling, but air passage is arranged by raising the glass-holding sash slightly, so as to permit a narrow invisible crack.

The 1931 west wing has many of its galleries unfinished, without any ceilings at present. If unfinished areas were left unheated, the rooms under them and alongside of them would require extra heating surface and there would be danger of condensation, freezing, etc.

The upward plan of heating the galleries, therefore, had advantages in meeting this situation, since with the well-distributed slot-inlets along the floors it makes no particular difference whether any gallery has even a roof over it, so far as getting that gallery warm is concerned.

The heating and air conditioning system as installed is essentially a recirculating system in which the air is tempered, moistened, dust-filtered, and drawn through supply fans in the basement. These fans discharge it through metal ducts to local reheaters in the basement directly under the rooms which they serve. The air for the ground story is exhausted back into the basement and is carried in metal ducts to the exhaust fans, which may waste it outboard or may return it to the separate supply systems of the ground story. The air supply for the galleries enters them at the floor, passes through the ceilings to the attic, and is drawn across the attic to down-going shafts leading to the supply fans, or is discharged outboard in proportions as convenient. The cold air intakes are from towers as high as possible above the dust-eddies which usually occur on flat city roofs.

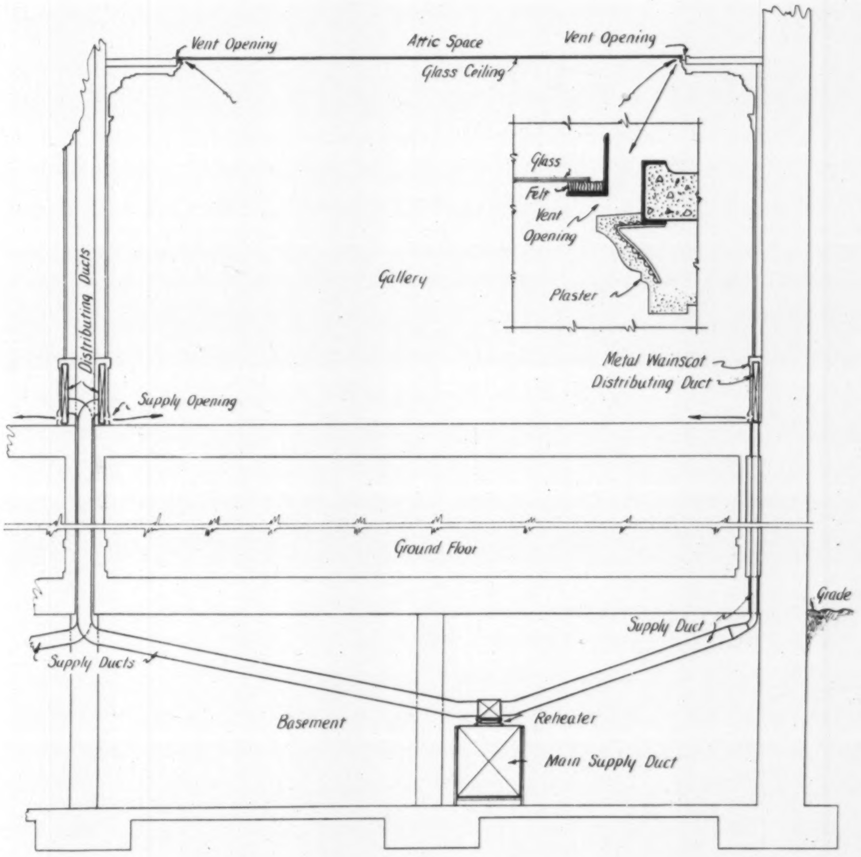
Walls and Roof Insulated

The outer walls and roof of the building are insulated with cork to reduce the heat loss when using the attic as a recirculating duct, since specific attic air ducts could not be installed for want of room. The circulation of the return air or the spent air across the attics has an advantage economically, in that no skylight heater coils are necessary.

The heat loss from dry air at about 70 degrees on its way to be recirculated totals a credit against the investment and operating cost of condensation-preventing radiators under skylights, such as are conventionally installed in many buildings of this type.

Air to Offices Supplied at Window Stools

The ground story rooms, in general, are used for classrooms and for administrative purposes. They are supplied with air at the window stools, through uplooking grilles from reheaters in the basement. The air is exhausted to ducts in the basement through vent registers at the inner ends of the classrooms at both the floor and the ceiling and which may be used alternatively or together.

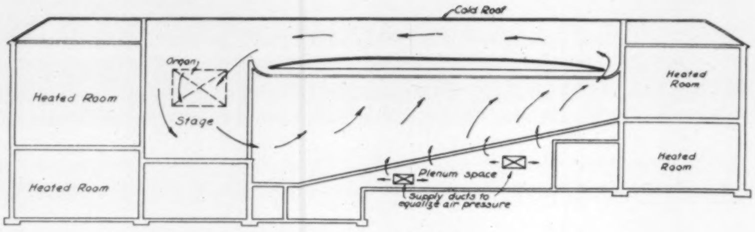


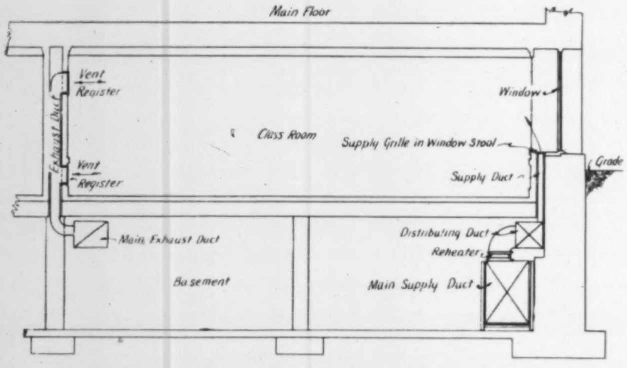
VERTICAL SECTION THROUGH A 1931 GALLERY

The air for each bay is heated by reheaters in the basement and enters the gallery through slots in the baseboard, which are practically continuous. It rises evenly over the area of the room and escapes to the attic. This arrangement is effective in winter and summer.

RIGHT — VERTICAL SECTION THROUGH THE 1923 HEMICYCLE

The roof-chill caused an objectionable interior air-circuit which was corrected by adding radiation over and alongside the stage.





LEFT—SECTION THROUGH A 1923 CLASSROOM

The vent openings near the ceiling are used in warm weather while those near the floor are used in cold weather. The air supply, heated in the basement, enters at the window stools.

Roof Is Ceiling of Music Hall

The 1923 hemicycle or auditorium and the 1931 music hall are similar in construction and different only in size. They have plain curved walls and "inverted platter" ceilings painted to simulate the sky, with very large open areas around the edges of the "sky"—to such an extent that the real ceiling is the roof of the building. The inverted platter was not considered as a ceiling, the roof itself being insulated. Upward, well-distributed ventilation with a low-velocity adjustable inlet under every seat is used and this scheme for this purpose has been very satisfactory. The plenum chambers under the auditoriums are zoned, with distributing ducts to equalize the air pressure and delivery, and there are separate reheaters, controlled by separate thermostats, for various zones.

Both auditoriums have separated supply and exhaust fans and are arranged so that during periods of non-occupancy they will keep fairly warm by gravity air recirculation. Both auditoriums are capable of being supplied efficiently with any percentage of outside air desired.

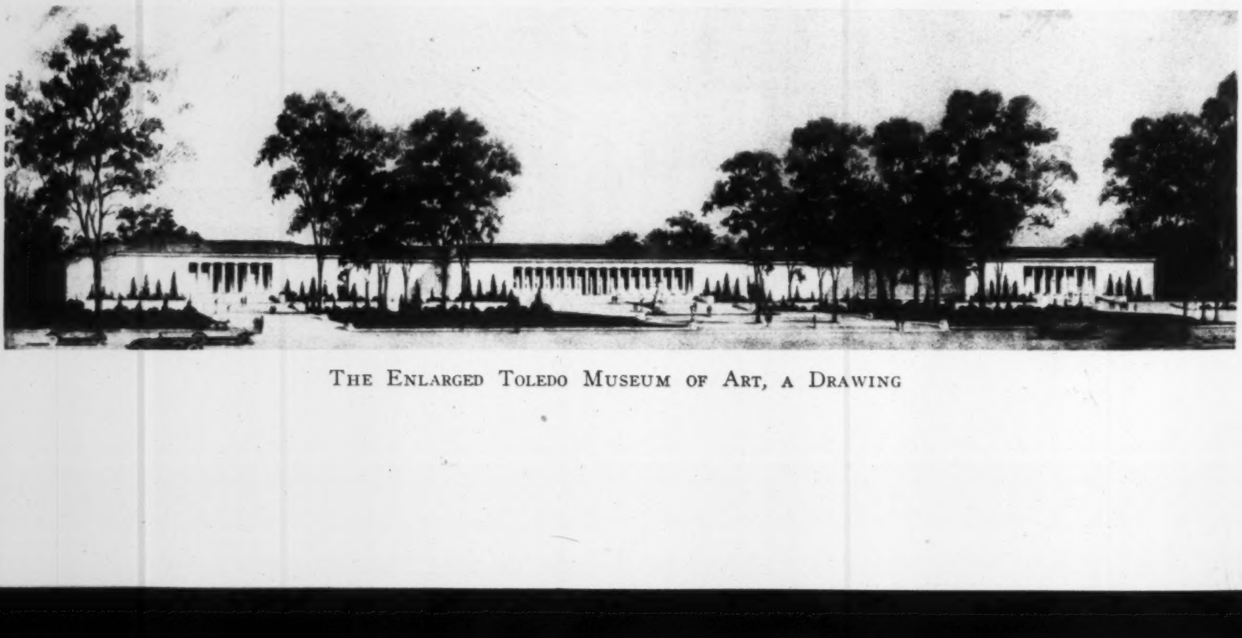
All toilet rooms, locker rooms, etc., have separate exhaust fans, and take their air supply from the adjacent corridors, so that by no chance can there ever exist a counter-current of air from the toilet room to the corridor.

Unit Heaters for Stage

The stage of the music hall has a very large cold wall and ceiling area. The walls and ceiling of the stage were insulated with cork, and six large unit heaters in three vertical zones are provided for heating this lofty room, in addition to strategically-placed direct radiators.

An interesting experience was encountered with the hemicycle in the 1923 center building, where the stage has no outside walls, its only cooling surface being its roof. In order to facilitate a very fine organ installation the upper part of the stage is wide open into the attic above the inverted platter or "sky" over the auditorium. No particular attention was paid, in the design, to the heating of the stage beyond the introduction from the auditorium plenum chamber of enough warm air near the floor on the stage side of the proscenium arch to compensate for the cool ceiling. As long as the fans were running everything was satisfactory, but as soon as the fans stopped the air, cooled by contact with the auditorium roof, found it more convenient to fall down through the stage like a mountain torrent of water than to diffuse evenly throughout the room, or to escape through the high-up exhaust opening. A very definite and unpleasant ring-circuit always established itself in cold weather as soon as the supply fans ceased running.

The cold air falling down in this way was of no use to the organ and the organ began to protest, until we were compelled to install something like 900 square feet of direct radiation in the upper part of the stage to end this complaint.



THE ENLARGED TOLEDO MUSEUM OF ART, A DRAWING

Hot Water Heating System Zoned

The building is heated by forced circulation hot water from the plant of the Rail-Light Company. It enters at a temperature regulated at the power house according to the weather; a differential pressure of about 25 pounds is available.

The entire hot water heating system is designed for positive circulation, using small size piping, with automatic temperature regulation of the water flow to the individual radiators. Such an arrangement is in itself an automatic adjuster of the water flow rate, since when the first rooms to get warm have their water shut off by the automatic radiator valves, that water builds up pressure which causes a more rapid flow in the cooler rooms, and as these, one by one, reach the desired temperature, the process becomes cumulative. In addition to this individual automatic control, however, the hot water mains are zoned for the major exposures and have electric master thermostats which accomplish general water temperature regulation. Each zone has a water pressure regulator on the flow main with its governor on the return main so adjusted that a constant differential in pressure between the two always is in effect; this differential pressure having been established by actual experience as that necessary to overcome the friction of the circuit. Thus none of the zones ever may take too much of the hot water and any one of them which has its pilot thermostat sufficiently warm will be cut down on its water supply volume and thus will have a reduced local differential pressure which will tend to cause a higher pressure for the remaining zones.

For the 1931 additions hard drawn copper tubing, solder-sweated forged brass fittings and special valves for use with copper tubing were used on all heating services.

Skylights Whitewashed in Summer

In order to reduce the heat gain due to direct sunshine in summer on the glass skylights over the galleries these roofs are whitewashed every summer with a lime solution which has been proved by cut and try methods to endure against the summer rains only; when fall comes, with its shorter days and reduced sun glare, Nature has cleared away the whitewash. Very little human aid is needed to restore the transparency of the glass when desired. Very great gains in comfort are possible when the roofs and skylights are thus made into reflectors rather than into absorbers of sunshine.

Turning Blades Reduce Duct Elbow Losses

By F. E. Wertheim

A CELEBRATED engineer once wrote a series of articles entitled "Things that are usually wrong." Inasmuch as he did not mention duct elbows, the subject can hardly be said to be exhausted.

Importance of Proper Duct Design

The apparent simplicity of the problem of designing ducts may possibly be the main reason for the numerous installations made without regard for the natural laws of air flow. The large volumes handled by ducts make the subject of duct losses one of considerable importance.

The air conditioning and ventilating engineer is not the only one concerned with the problem. For example, a power plant equipped with large economizers and air preheaters and having fan motors, perhaps as large as 500 hp., may waste a considerable percentage of auxiliary power if the usual extensive system of ducts has been designed incorrectly. Redesign, involving only moderate expense, may effect both an increase in station capacity and economy in the auxiliary power consumed by fan motors.

The engineer must bear in mind that "an air conditioning installation is no better than its duct or distributing system." Let us imagine some plant in which there is a leaky oil tank having a visible waste of several barrels of oil per day—a leak of this sort would be speedily repaired. On the other hand, the benefits to be realized by stopping the insidious power leaks due to resistance in poorly designed elbows in ventilating systems have often been ignored.

To insure maximum effectiveness of conditioned air, duct work should be designed for proper distribution with minimum operating cost, i.e., power cost of the fan.

References Cited

The purpose of this article is to outline briefly the requirements for good design, so that the engineer can lay out elbows which will reduce duct losses to the minimum. Manifestly, an abstract of preceding contributions to the technical press is necessary. The writer has, however, endeavored to avoid a mere "re-write" of the original work given in the appended bibliography of several important articles.

Attention to Duct Adjacent to Elbow

Study of elbow losses involves attention to the design of the adjacent duct. It hardly seems necessary to state that the optimum duct is of circular cross-section, or that a square section is preferable for straight runs whenever round sections are not to be used. Entrance losses may be reduced by the shape of inlets, the proper design of the elbow itself will reduce turbulence and eddy losses, and whenever feasible a straight pipe at least four diameters in length should be placed at the elbow exit to smooth out the flow, with resulting lessening of losses. The straight pipe at exit will bring about the same gains for either round or rectangular sections.

Typical Elbows

Referring to Fig. 1, the ratio W/D is called the aspect ratio, and R/D the radius ratio. In general, elbows should be designed with large radius ratios and large aspect ratios. There is no gain to be had, however, in making either ratio greater than six.

Fig. 2 shows a conventional elbow which may be improved in several ways. Fig. 3, showing a square corner elbow, does not appear in very many illustrations of

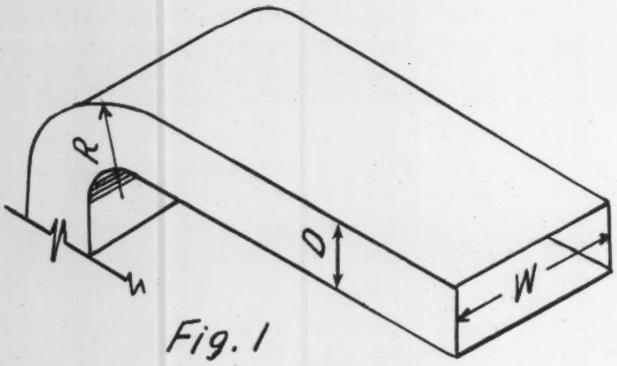


Fig. 1

FIG. 1—THE ASPECT RATIO OF AN ELBOW IS W/D. THE RADIUS RATIO IS R/D

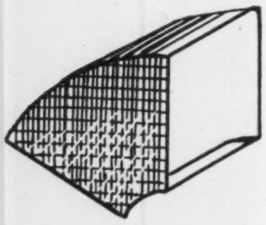


Fig. 2

FIG. 2—A COMMON TYPE OF ELBOW WHICH OFTEN CAN BE IMPROVED

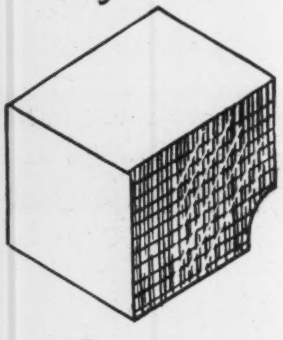


Fig. 3

FIG. 3—THIS SQUARE CORNER ELBOW HAS TEN PER CENT LESS LOSS THAN THE ELBOW SHOWN IN FIG. 2, ACCORDING TO TESTS

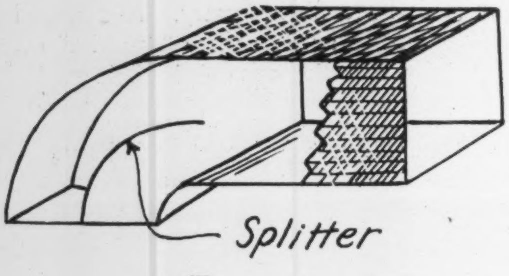


Fig. 4

FIG. 4—ROUNDED ELBOW TO WHICH A SPLITTER HAS BEEN ADDED, INCREASING THE ASPECT AND RADIUS RATIOS

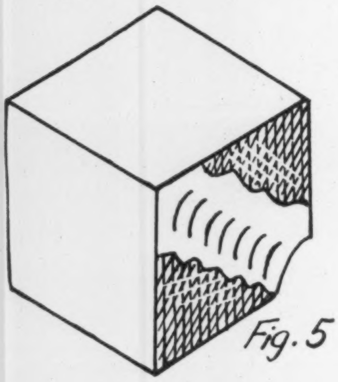


Fig. 5

FIG. 5—THE ADDITION OF TURNING BLADES WILL OFTEN REDUCE LOSSES

duct work. Nevertheless this square corner elbow has 10 per cent less loss than the round corner elbow in Fig. 2, according to tests.

Fig. 4 shows a rounded elbow to which a splitter has been added, thus increasing both aspect and radius ratios, with resultant economy. Two splitters should be the maximum.

Structural conditions may impose the requirement of a square bend with an aspect ratio of less than unity and a very sharp bend, giving a flat turn with extreme loss of velocity head. The addition of turning blades, as shown in Fig. 5, may reduce losses as much as 80 per cent. The chart, Fig. 6, is intended for use in designing square elbows equipped with blades; it is based on an aspect ratio of 4; a radius ratio of 1.25; and length of lip equal to 0.6 × radius.

The edges of the blades, made of sheet metal, should be thinned to half the thickness of the metal, preferably by a gradual taper (7° or less).

Bibliography

New Data for the Design of Elbows in Duct Sys-

Important power savings can be made by reducing duct losses to the minimum. No plant would allow a leaky oil tank to waste several barrels of expensive oil every day, yet often improperly designed ducts and elbows are allowed to waste just as expensive power month after month. Frequently, turning blades installed in the elbows will lead to decreased resistance and greater efficiency. The chart published with this article gives the necessary information for their use.

tems, by Loring Wirt. (*G. E. Review,* June, 1927. Vol. 30; No. 6.)

Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey. (*Transactions, A. S. H. V. E.* Vol. XIX, 1913, p. 366.)

Change in Duct Design Saves 400,000 kw. per Year, by E. D. Edwards. (*Power,* Oct. 23, 1928.)

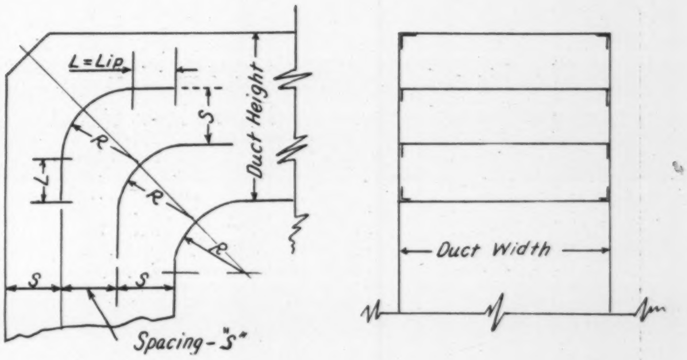
Square Bends Reduce Elbow Losses in Ducts, by B. L. Spivak. (*Power,* Jan. 8, 1929.)

Reducing Friction Losses in Rectangular Duct Elbows, by H. Nippert. (*Power,* June 10, 1930.)

Directions for Using Chart

From horizontal scale of *Duct Height* enter chart on vertical line to intersection of *Duct Width* on left vertical scale; thence vertically to nearest inclined N-line to obtain number of blades in elbow.

From point of intersection of *Duct Height* and N-line, follow horizontal line to intersect two right hand scales to obtain radius R and lip L; and vertical S-scale to obtain spacing of blades.



Example: Given a duct 7 ft. 6 in. high by 9 ft. 3 in. wide.

Find . . .
(a) Number of blades
(b) Spacing
(c) Radius of blades
(d) Length of lip

(a) N = 2 = number of blades
(b) S = 30 in. = spacing
(c) R = 37½ in. = radius
(d) L = 22½ in. = length of lip

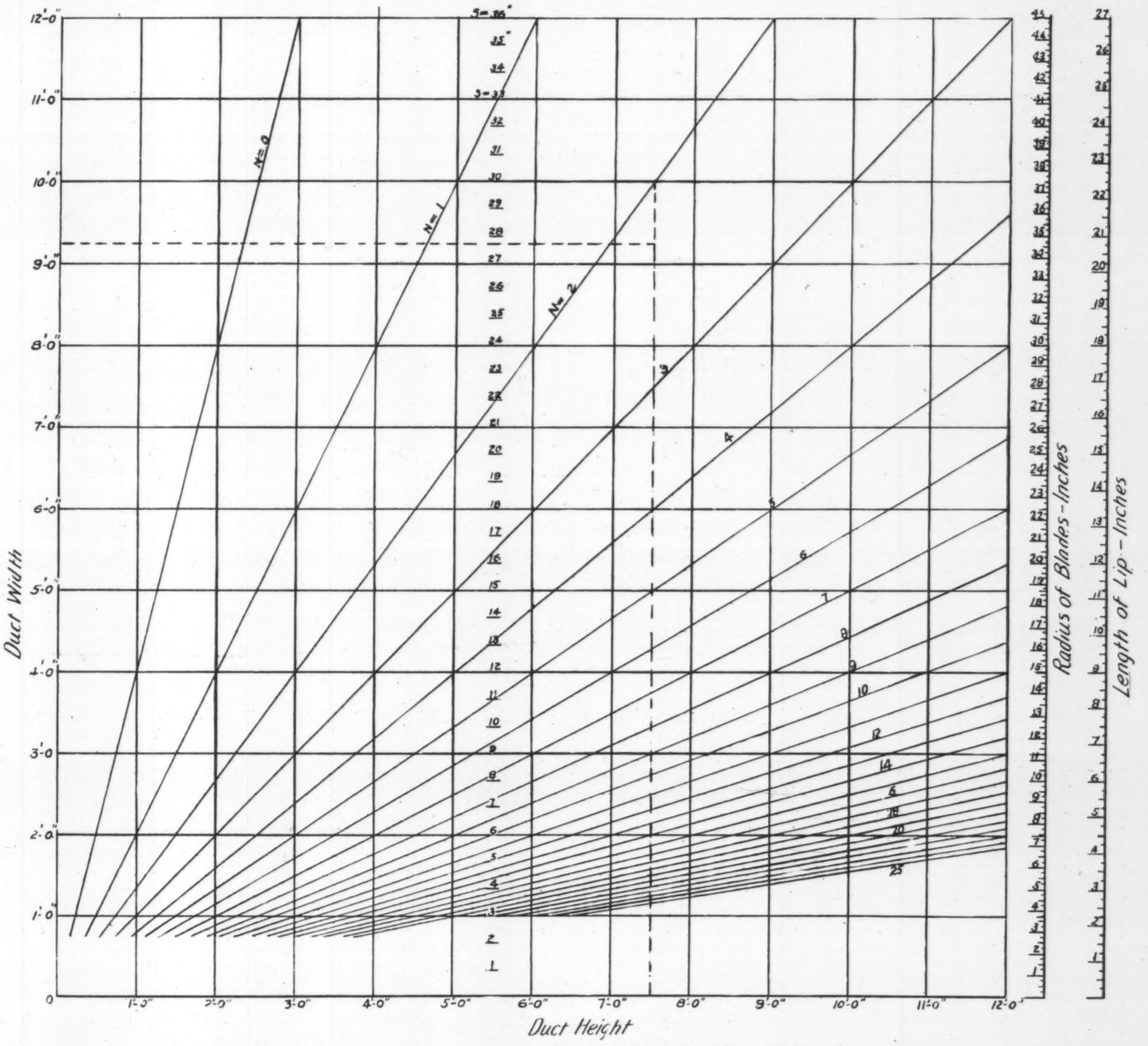
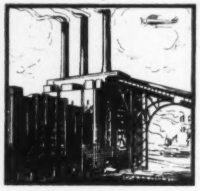


FIG. 6—CHART FOR DESIGNING SQUARE ELBOWS EQUIPPED WITH TURNING BLADES

In this article, the author considers the design of joints for piping 1000-F steam at the Delray plant of The Detroit Edison Company. Part I of this article—which was published last month—related to the flexibility of high temperature piping

By Arthur McCutchan*



Designing High Temperature Steam Piping

Part II—Pipe Joints for 1000 F Service

A STUDY of bolted gasket joints for operation at temperatures from 1000 F to 1100 F indicates that it is expensive to design a rigid type of joint for high working pressures at this temperature due to the effect of creep on such a joint.

These considerations led to the use of welded joints backed up with flanges for the joints where disassembly was not required. Four joints in the high temperature line from the superheater to the 10,000 kw. turbine[1] were desired which could be taken apart readily. Two joints were made of extremely heavy construction of modified 900-lb. *A.S.A.* dimensions while the remaining two joints, whose dimensions were fixed by previous design, were provided with springs to compensate for the effects of creep.

Flexibility of A.S.A. Flanged Joints

Flanged joints made up to *A.S.A.* standard dimensions are found to possess considerable flexibility. In general the deflection of the flanges in a loose-flanged joint will be from two to four times as large as the change in length of the bolts for a given change in load on the joint. This change in load on the joint may be due to temperature lag of the bolts behind that of the inner edge of the flange in bringing the line up to

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*Engineer, Engineering Division of The Detroit Edison Company, Detroit, Mich.

[1] See description in Part I of this article. (Page 825, HEATING, PIPING AND AIR CONDITIONING, October, 1931.)

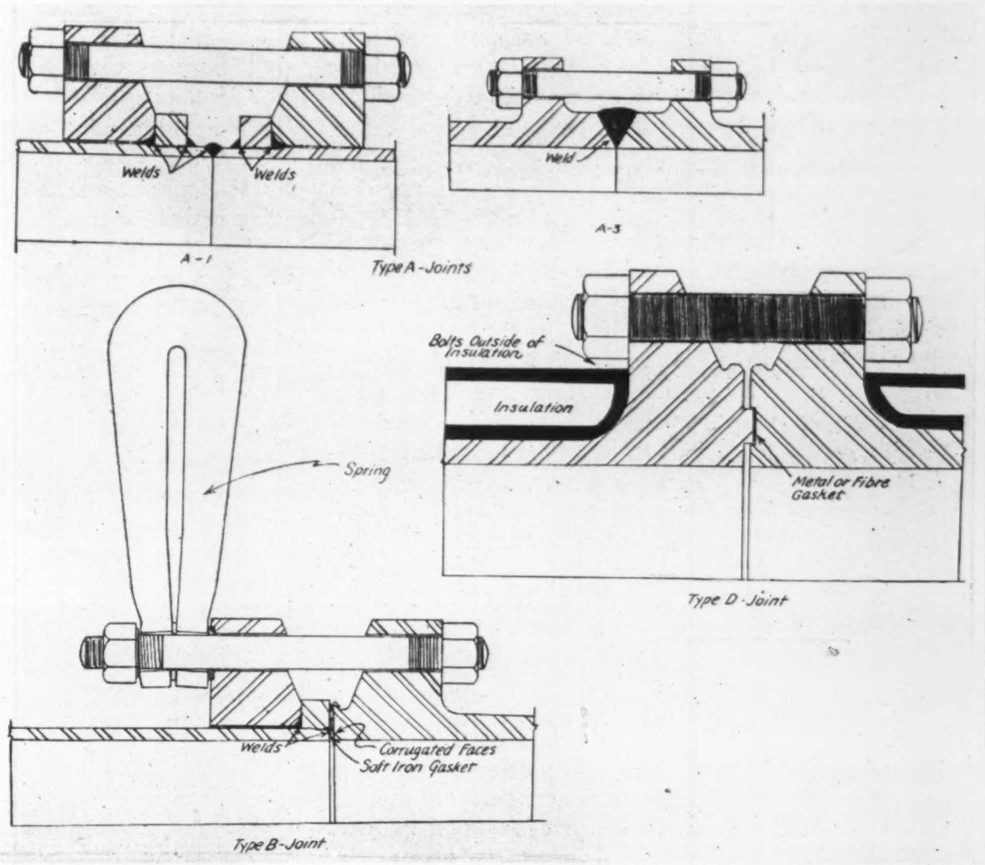


Fig. 2—Various Types of Pipe Joints Used in the 1000-F Piping at Delray

operating temperature or may be caused by *creep* or gradual accommodation of the flange and bolt material to the stresses which exist in the joint in the operating condition.

An exaggerated picture of the effect of temperature difference on the expansion of a flange is shown in Fig. 1. The difference in expansion which tends to set up additional stress in the bolts is caused by the change in length of the mass at the inner edge between the center of the flange and the gasket surface acting at the higher temperature and the mass between the center of the flange and the nut (Mass 2) acting at a lower temperature. The difference between the sum of these elongations and that of the bolt over the corresponding length is one-half the amount that the bolt will be stretched because of the temperature lag. Published data from tests indicate that the maximum temperature difference during the warming-up period which might be expected in a fully insulated flange when the steam temperature is raised to 1100 F should not exceed approximately 300 F.

If the flanges were perfectly rigid such a temperature difference would set up extremely high stresses in the bolts and flanges, but—as noted in the foregoing—ordinary *A.S.A.* flanges actually possess considerable flexibility. This flexibility of the flanges reduces the stresses set up by temperature difference during the warming up period to from one-half to one-fourth that which would exist if the flanges did not deflect. Neglect of the flexibility of the flanges is responsible for erroneous conclusions which have been drawn as to the magnitude of bolt stresses caused by temperature difference. The flexibility of the flanges likewise tends to minimize the effect of creep and hence extends the life of the joint.

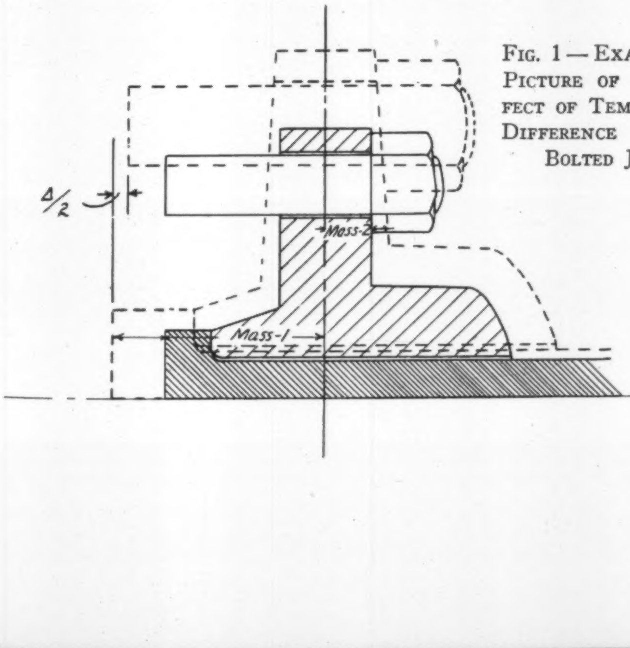


Fig. 1 — Exaggerated Picture of the Effect of Temperature Difference on a Bolted Joint

Design of Heavy Flanged Joint

It is possible that, if a gasket surface is used which requires a small axial force to keep tight, and if extremely heavy bolting and flanges are used so that the working stresses are very low, the creep of the flange and bolt material may be compensated for by periodic tightening of the nuts without introducing supplementary flexibility into the joint.

Creep Characteristics of 10-in., 600-lb. Flanges

However, in the case of the 10-in., 600-lb. *A.S.A.* flanges of the center-pressure valve which connects to the outlet tee of the superheater, the working stresses are such that rather high creep rates may be expected in the flange and bolting material. As there have been no creep tests made on flanges, or even on beams subjected to bending, only qualitative results can be obtained from an analysis of the creep characteristics of a joint. Hence in this study only the creep of the bolts is taken into account but the creep curves are chosen so that an allowance exists both because of the use of higher grade bolting material and the use of bolts with full shanks in the final design instead of bolts threaded the entire length as were used in determining the relative life of joints in this study.

Development of Spring to Absorb Creep

As the result of an investigation of several types of spring or other flexible member, it has been determined that a simple and effective method of introducing flexibility into an existing joint is by the use of so-called "hairpin" springs, as shown in Fig. 2 at *B*. These springs are designed to approximate beams of uniform strength, so are economical of material. A high grade spring material with an allowable working stress of 60,000 lb. per sq. in. is used so that the ratio of effectiveness to weight is low. It has been found from temperature measurement on some springs installed in an experimental superheater line at the Trenton Channel plant that the maximum temperature of the springs does not exceed 500 F when the steam temperature is 1100 F. Consequently, the spring material can be assumed to retain its high physical properties under operating conditions.

Effect of Spring on Life of Joint

Regardless of inaccuracies in the assumption as to creep rates, effect of bending moments, operating temperatures, etc., the use of these springs should increase the life of the particular joints to from ten to twelve times the period which they could be expected to stay tight without their use. This is indicated in Fig. 3. For example, if the length of the bolt is changed by 0.030 in. either due to creep or temperature difference the load on the individual bolt will be changed by 9,800 lb. as found from the load deflection relation of the joint without a spring. To obtain this same change in load on the bolt in a joint with these springs installed would require a change of 0.37 in. in the length of the bolt. In other words if this change in length were due to creep it would require more than twelve times as much creep to effect the same reduction in bolt load.

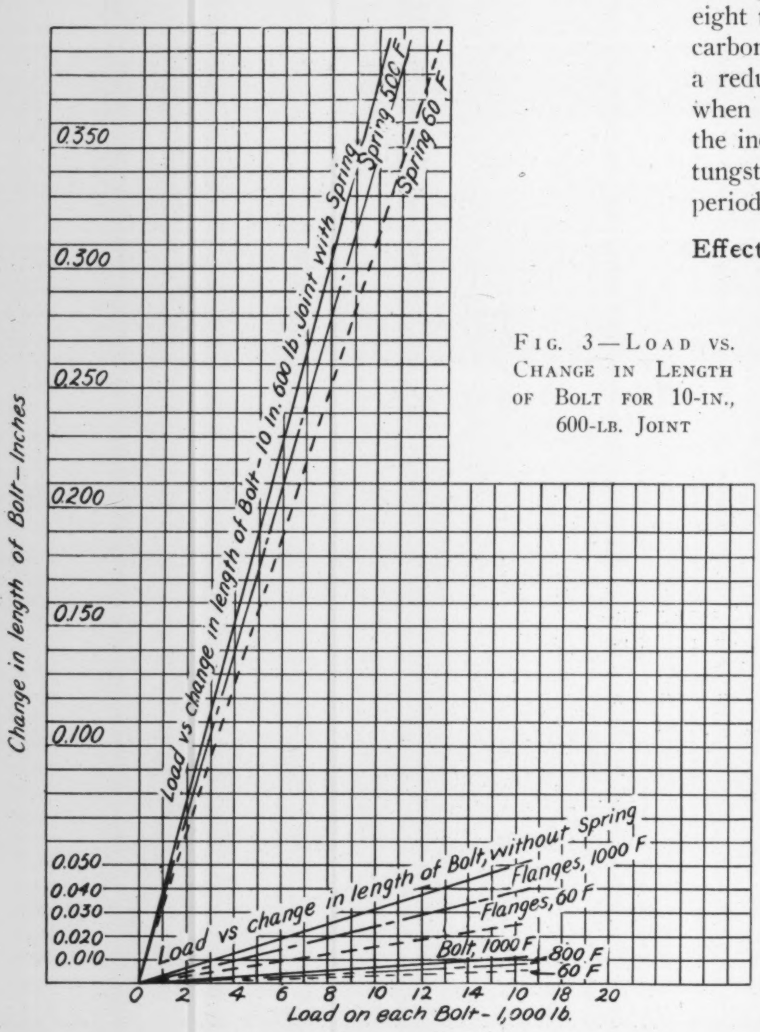


Fig. 3—Load vs. Change in Length of Bolt for 10-in., 600-lb. Joint

The foregoing relation is given more directly in Fig. 4 in which the gasket compression on the side of a joint, which has its gasket pressure reduced by a bending moment of 140,000 in.-lb., is plotted against operating period in 1,000 hours. If it is assumed that the joint will leak when the gasket compression is reduced to slightly below twice the W.S.P. (1¾ in Fig. 4) then the use of springs increases the life of the joint from 250 hours to 4,000 hours, or sixteen times. If the joint remains tight until the gasket compression is reduced to once the W.S.P., the joint life is increased from 500 to 6,000 hours, or twelve times. These results are based on creep data of Professor F. H. Norton for 0.37C steel.

If creep data for a chrome-tungsten bolt steel similar to that selected for this installation are used as a basis, the operating periods are found to be approximately eight times as long as those based on Norton's data for carbon steel, but the increase in length of time to effect a reduction of gasket compression to once the W.S.P. when springs are used does not differ materially from the increase found for carbon steel. With the chrome-tungsten steel, the joint will remain tight 12.8 times the period which would elapse if springs were not used.

Effect of Spring on Bolt Stress Due to Temperature Difference

The effect of temperature difference on bolt stress during the warming-up period is shown in Fig. 5. It is assumed that the bolts have attained 800 F when the flange mass at the inner edge is at 1050 F. A bending moment of 140,000 in.-lb. increases the stress in the bolts in one-half of the flange and decreases the stress by a like amount on the other side. Bolt forces, bending couples, and gasket reactions are considered as acting at the centers of gravity of their respective semi-circumferences.

The drop in the modulus of elasticity with increase in temperature is responsible for the reduction in bolt stress at operating temperature. Without the use of springs and an initial tension cold of 10,000 lb. per sq. in., the maximum stress attained during the warming-up period is approximately 17,500 lb. The use of the springs eliminates this factor from consideration as the increase in bolt tension due to temperature difference is compensated for by the deflection of the spring and the drop in value of the modulus of elasticity. (See Fig.

6.) Fig. 5 also shows the reduction in bolt tension due to creep during the first month or two of operation.

Bolts with Full Shanks vs. Bolts Threaded Entire Length

From the standpoint of reducing stress due to temperature difference the use of bolts threaded the entire length or with the shanks reduced to the root diameter of the thread can be justified with certain types of joint. If the flanges of the 8-in. joint were perfectly rigid and were connected with twelve 1½ in. bolts having an unthreaded portion twelve inches long and three inches of threaded length between the centers of the nuts, the elongation of the bolts in terms of the total axial load on the joint and the modulus of elasticity would be given by:

$$\delta_1 = \frac{12P}{21.2E} + \frac{3P}{16.86E} = 0.745\,\frac{P}{E}$$

The similar expression for the same bolts threaded the entire length is

$$\delta_2 = \frac{15P}{16.86E} = 0.890\,\frac{P}{E}$$

If, due to temperature difference, the distance between the centers of the nuts is increased by 0.015 in., the resulting increase in axial load on the joint will be:

Full shank bolt

$$P_1 = \frac{0.015 \times 24{,}000{,}000}{0.745} = 483{,}000 \text{ lb.}$$

Threaded bolt

$$P_2 = \frac{0.015 \times 24{,}000{,}000}{0.890} = 404{,}000 \text{ lb.}$$

The total cross-sectional area of the bolts in the joint at the roots of the threads is 16.86 sq. in. The increase in stress due to temperature difference for the full shank bolt is 28,700 lb. per sq. in. The similar figure for the threaded bolt is 24,000 lb. per sq. in. In other words, the use of full shank bolts results in an increase in stress due to temperature difference of approximately 20 per cent in a hypothetical case in which rigid flanges are assumed.

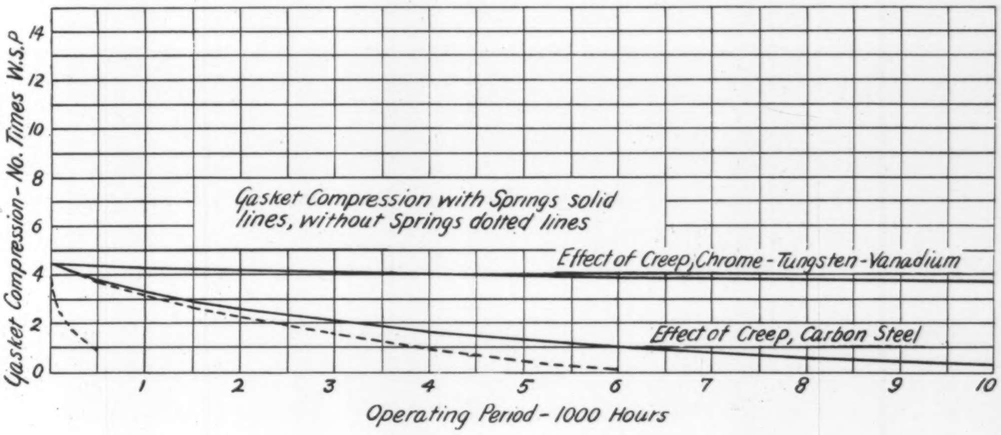


FIG. 4 (ABOVE)—GASKET COMPRESSION ON SIDE OF JOINT WHICH HAS PRESSURE DECREASED BY BENDING MOMENT

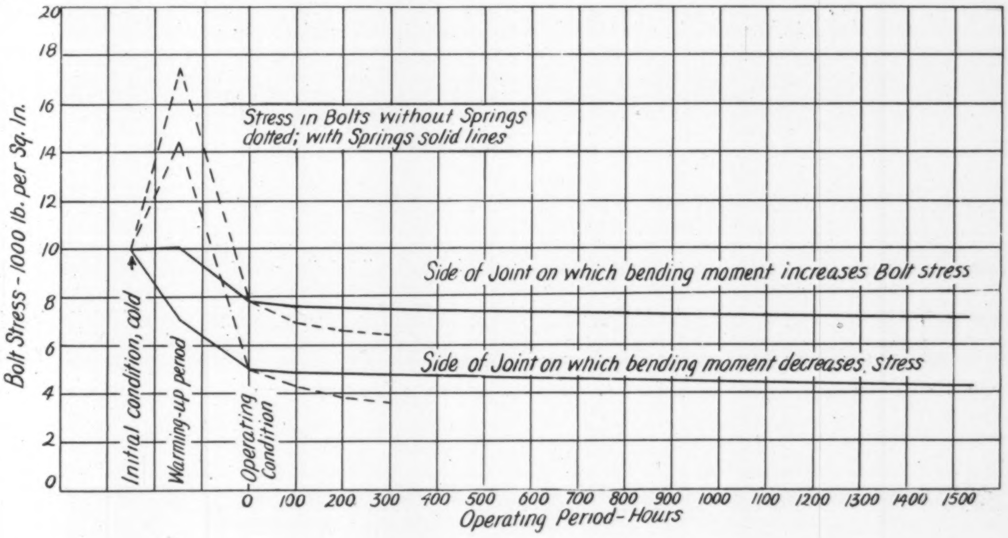


FIG. 5—STRESS IN BOLTS, MODIFIED 10-IN., 600-LB. *A. S. A.* FLANGED JOINT

However, in the particular 8-in. flanged joint under consideration, the deflection of the flanges for a given load is more than four times the elongation of the bolts and is given by

$$\delta_3 = \frac{3.93\,P}{E}$$

Hence the expressions for elongation of the bolts under the two conditions become

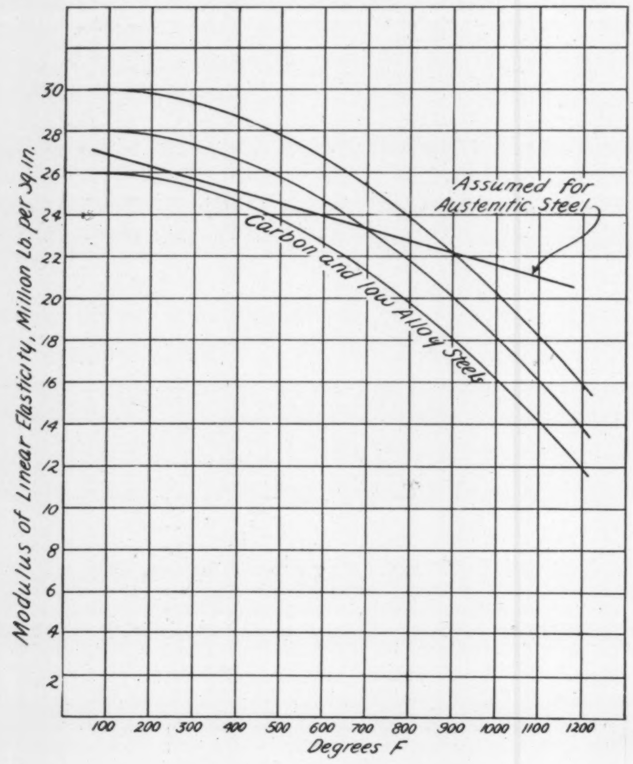


FIG. 6—REDUCTION IN MODULUS OF ELASTICITY WITH INCREASE IN TEMPERATURE

$$\delta_1 = 0.745\frac{P}{E} + 3.93\frac{P}{E} = 4.675\frac{P}{E}$$

and

$$\delta_2 = 0.890\frac{P}{E} + 3.93\frac{P}{E} = 4.820\frac{P}{E}$$

The increase in stress due to the same temperature difference as used in the case of rigid flanges shows the following relative stresses:

Full shank bolt

$$S_1 = \frac{P_1}{A} = \frac{0.015 \times 24{,}000{,}000}{4.675 \times 16.86} = 4{,}570 \text{ lb. per sq. in.}$$

Threaded bolt

$$S_2 = \frac{P_2}{A} = \frac{0.015 \times 24{,}000{,}000}{4.820 \times 16.86} = 4{,}430 \text{ lb. per sq. in.}$$

In this case, where the flexibility of the flanges is taken into account, it appears that the use of full shank bolts results in an increase in stress due to temperature differance of only three per cent.

It is thus apparent that the use of full threaded bolts can be justified only in the case of very rigid flanged connections and where the increase in stress due to temperature difference approaches the elastic limit of the bolt material. Furthermore, full threaded bolts can be justified only where the operating temperature is such that creep does not complicate the problem.

For operating temperatures in the neighborhood of 1000 F, creep of the material becomes the important factor in joint design. Under such conditions the use of full shank bolts is preferable due to the reduction in length of bolt subjected to high stress and consequent higher rate of creep.

This advantage of the use of full shank bolts under creep conditions may be illustrated by assuming that the bolts in the foregoing examples are tightened so that

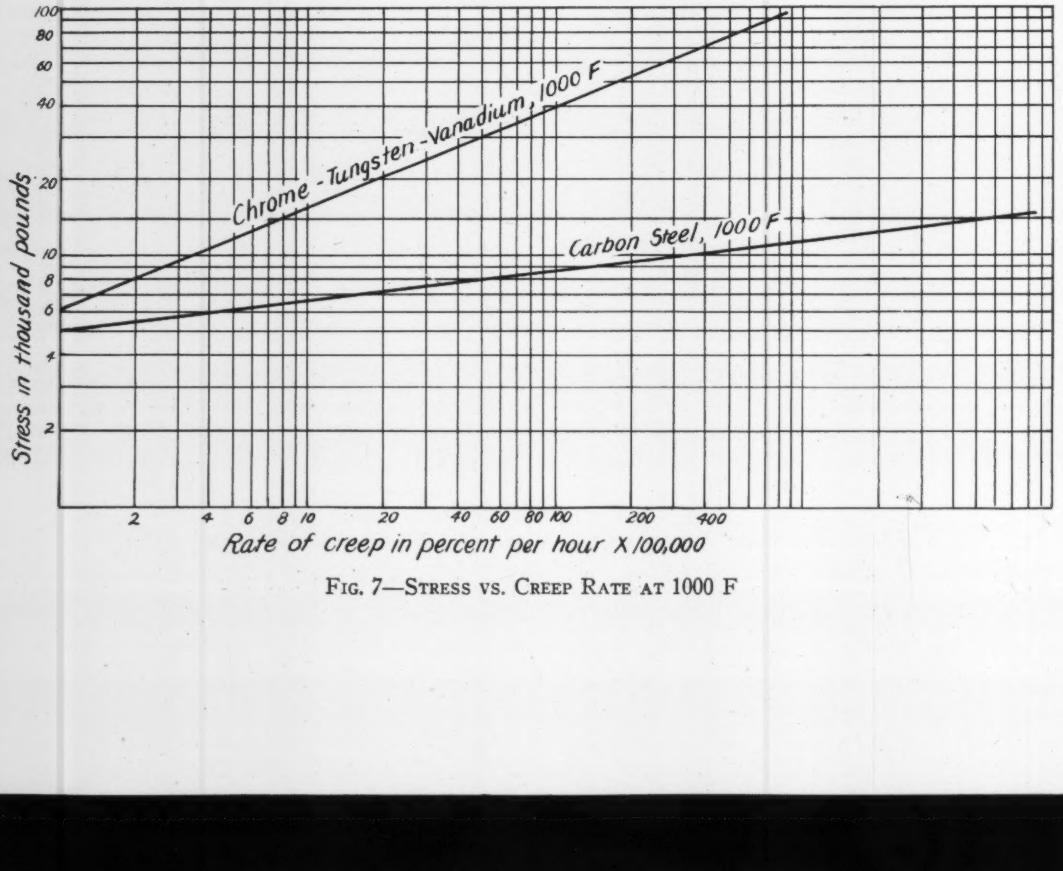


FIG. 7—STRESS VS. CREEP RATE AT 1000 F

the total axial load on the joint when temperature equilibrium is established is 168,600 lb., giving a stress in the threaded portion of both sets of bolts of 10,000 lb. per sq. in. The stress in the unthreaded part of the full shank bolts will be in the ratio of their total cross-sectional areas, or:

$$\frac{16.86}{21.2} \times 10{,}000 = 8{,}000 \text{ lb. per sq. in.}$$

From the creep curve for a chrome-tungsten steel (Fig. 7) the rate of creep for a stress of 10,000 lb. per sq. in. is found to be 3.5 per cent in 100,000 hours. The similar figure for 8,000 lb. stress is 2 per cent in 100,000 hours.

For the full shank bolts the increase in length in 10,000 hours due to creep, assuming constant stress, would be

$$12 \times 0.002 + 3 \times 0.0035 = 0.0345 \text{ in.}$$

The bolts threaded the entire length would creep under similar conditions an amount given by

$$15 \times 0.0035 = 0.0525 \text{ in.}$$

In this particular comparison the total increase in length of the full shank bolts due to creep in 10,000 hours is only approximately 65 per cent as large as in the case of the bolts threaded the entire length. Of course, the reduction in stress as creep occurs must be taken into account in determining the life of a joint, as was done in determining the effectiveness of the springs; the relative advantage of full shank bolts vs. threaded bolts will depend also upon the slope of the creep curves employed, but the general conclusion that full shank bolts should be used under temperature conditions involving creep will not be altered by these considerations.

Conclusions

The foregoing study indicates that for working temperature at or near 1000 F, flexibility in a bolted joint is essential if long life is to be obtained without employing excessively heavy construction. This flexibility may be obtained either by modification of the flanges or the introduction of some form of spring. The form of spring proposed enables a large increase in flexibility to be obtained in existing joints and should give a joint life at least ten times that possible without their use in this particular installation. The use of full shank bolts in these joints is indicated because of the reduction in creep and because the presence of springs renders unnecessary the slightly greater flexibility of full threaded bolts.

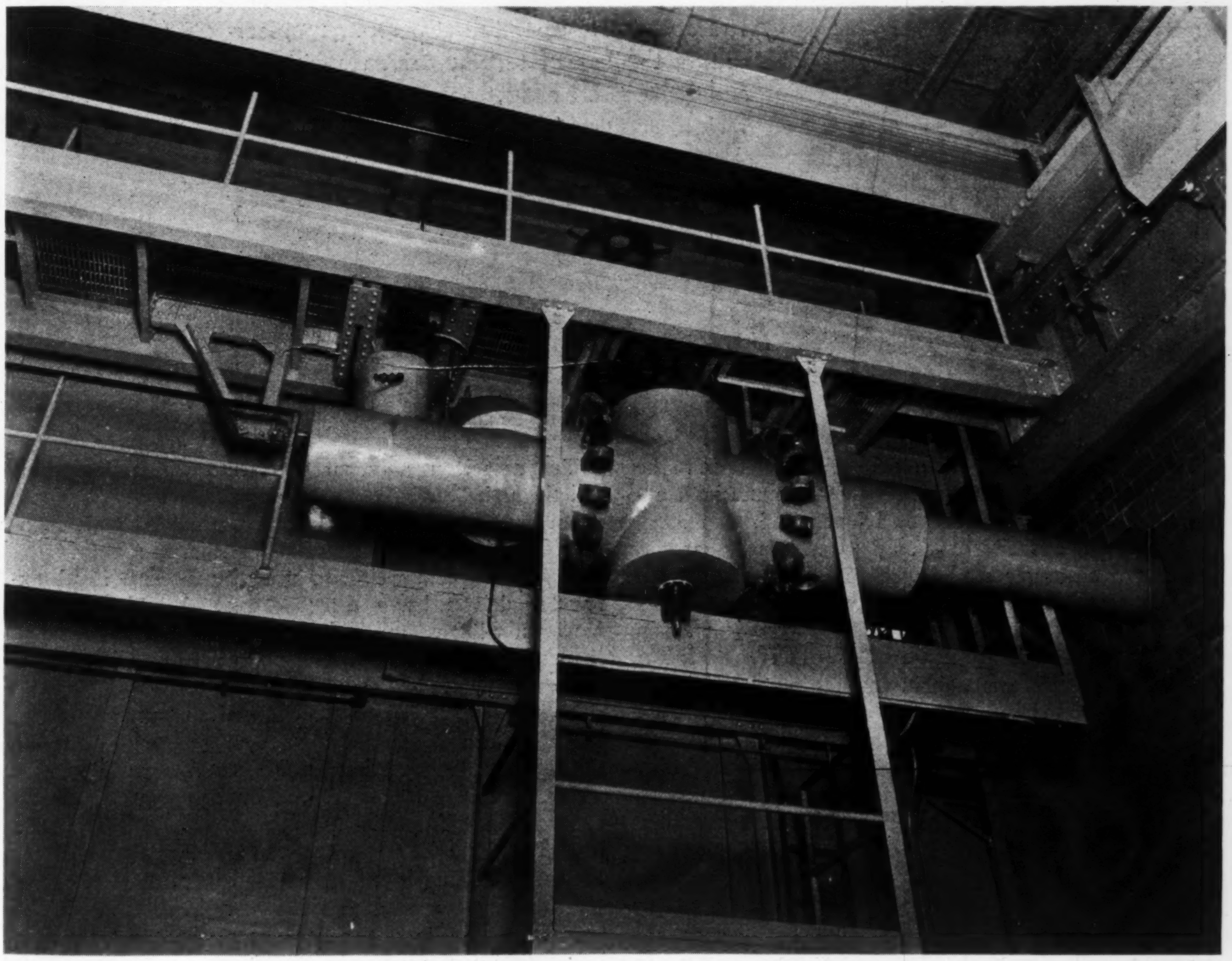


Fig. 8—Spring-compensated Flanged Joints—1000-F Stop Valve

Solution of the Ash-Handling Problem in a Large Office Building

This article describes the ash handling equipment installed in a large office building. The boiler plant supplies steam for direct and indirect radiation, for cooking and washing apparatus in three kitchens, and for tenants as required, including a barber shop and hat cleaning establishment. Hot water generators are also supplied with steam.

The boilers are operated at 80-lb. gage, permitting the use of turbine-driven auxiliaries, and the accurate reduction of pressures for the various requirements in a single stage.

Among the offices heated by this installation are the headquarters of the American Society of Heating and Ventilating Engineers.

By Henry G. Schaefer

DETERMINATION of the kind of fuel to be used in the boiler plant is often materially affected by the matter of ash removal. Good engineering demands that not only investment, depreciation, and operating costs be considered, but that in addition to these factors, the sanitary and space requirements of the structure be met. An interesting solution of this problem under extreme conditions is reflected in the ash-handling installation in the home office building of the New York Life Insurance Co. in New York City.

A 2,000 hp. chain grate stokered boiler plant burns No. 2 buckwheat coal. The firing floor is 60 ft. below street level, and the floor of the ash handling space 11 ft. below the firing level. The illustration diagrams the general arrangement of the vacuum type ash-handling equipment. Rack and pinion operated air tight gates at the bottom of the boiler hoppers *A* permit the ash to drop through the dust-preventing chutes *B* (which are hinged to the bridge wall foundation for convenience in cleaning). A screened air inlet *C* admits air to 8-in. chilled cast iron conveyor pipe, into which the ash is fed through "anvil" type openings *D,* which are closed when not in use. Cast iron plugs, provided with openings for a hand hook, are used for this purpose.

A 48-oz. vacuum is produced by a centrifugal exhauster driven by a direct connected 90-hp. electric motor at 3400 r.p.m. This vacuum is communicated to the dust separator *E,* the ash storage bin *F,* and the conveyor pipe line, successively.

The ash is carried on the induced air current up and into the storage bin. The conveyor pipe enters the bin tangentially. A cast iron strike plate is provided as a wearing surface for the initial impingement of the incoming ash. The velocity of both air and ash is soon dissipated by friction against the concrete lining of the bin (which lining is applied to the entire interior, except the top, over a coating of rust-resisting putty) and eventually all the ash, except some fine dust, falls to the bottom of the bin.

The exhaust connection is taken from the center of the top, so that the "core" of the air, which has discharged its centrifugally separated ash content and has lost most of its rotary motion, is drawn off, and then passes through the dust separator, where a second stage of centrifugal separation is offered for the dust carried over.

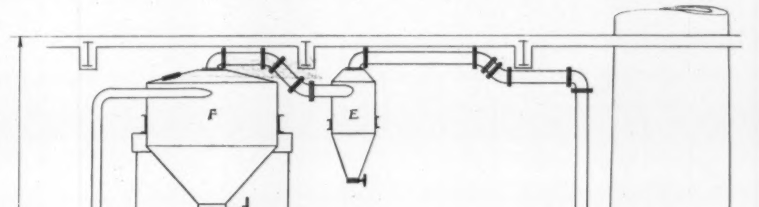
The discharge of the exhauster is carried into the main smoke breeching at its point of entry into the masonry stack. This connection is so arranged as to eliminate possibility of the exhausted air coming into contact with the hot breeching plates to cause rapid oxidation.

The complete elimination of noise was a major requirement in the design of this building's mechanical equipment. The exhauster is mounted on a concrete block set in, and isolated from, the walls and bottom of a concrete lined rectangular pit. Natural cork, 2 in. thick, is the isolation medium. The inlet and discharge pipe connections are broken by means of short sections of heavy rubber tubing, so that no metallic contact exists.

The supports of the entire piping system, except the main conveyor pipe under the ash hoppers, and including air piping and bins, are isolated from the building structure by either lead or hair-felt, or in some instances by a combination of both.

To facilitate dust separation, a water spray jet is provided in the conveyor at its point of entry to the ash bin. The valve controlling the jet is located beside the exhauster controller for convenience in operation. The water jet serves a dual purpose in that, since the ash is sprayed with water, its eventual discharge by gravity into trucks in unaccompanied by objectionable dust clouds.

It is interesting to know that this conveyor system is operated at will during office hours without the slightest hint of such activity being noticeable in even those public and working spaces through which the conveyor passes.



General Arrangement of the Vacuum Type Ash-Handling Equipment in the Home Office Building of the New York Life Insurance Co.

The Report to the Committee on Growth and Development, White House Conference on Child Health and Protection—Part I.

The Influence of Atmospheric Conditions on Health and Growth

● *By* C. P. Yaglou* ●

CLIMATIC influences upon the organism have been recognized since the time of Hippocrates, but our knowledge of the specific causative factors and of the physiological and biological principles involved is still very limited. Whatever the mechanism of action may be, it is now well known, from numerous statistical, laboratory and field researches at home and abroad, that atmospheric conditions in general, and temperature in particular, have a direct effect upon health, growth and activity, as well as indirect effects through food, disease, and mode of life.

Young children, especially infants under one year of age, are much more sensitive to the effects of heat and cold than adults, because their mechanisms for regulating the temperature of the body are less stable than those of adults. In no other age group is the incidence of morbidity and mortality so immediately and markedly affected by atmospheric conditions. The sensitivity decreases as the children grow older.

The whole subject of atmospheric influences is extremely involved and the literature lacks uniformity. An enormous amount of work is needed in all directions in order to segregate the effects of the various atmospheric factors and to gain an insight into the mechanism of their action, *first* upon man, *second* upon the organisms causing infectious diseases, and *third,* upon the means of transmission of the infectious agents. The present report deals largely with the effects of atmospheric conditions upon health and growth, as well as with methods suitable for the control of these conditions.

Part I—Climate and Season

Seasonal Variation in Morbidity and Mortality

The most conspicuous climatic influence upon the organism is the effect of season upon the morbidity and mortality of infants. In the temperate climates of the United States, seasonal changes occur principally in two groups of diseases, (a) diseases of the respiratory tract,[1] which run high in cold weather and (b) diseases of the intestinal tract,[2] which are prevalent in warm weather. Other seasonal diseases, not readily associated with these two specific groups, have declined in recent years without affecting decidedly the two seasonal peaks. Further north, the long and extremely cold winter constitutes the only unfavorable season, and further south the long hot summer is the only unfavorable season.

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[1] Including acute bronchitis, pneumonia of all forms and influenza.

[2] Including diarrhea and enteritis.

As a means for measuring health, sickness records are inadequate because the actual morbidity is not expressed in statistics. Accurate infant mortality rates, on the other hand, are available for the birth registration area of the United States, and these offer a very sensitive index of the incidence and severity of the two specific groups of seasonal diseases.

Fig. 1 shows the seasonal variation in infant mortality from these diseases in the United States birth registration area. It will be seen that the mortality from diseases of the respiratory tract reaches a minimum value in July and August, which are the warmest months in the year. With the onset of cool weather in September, the mortality from these causes rises, attaining peak values between January and March. The mortality from diarrhea and enteritis follows just the opposite course. It begins to rise early in May, reaching its maximum in August or September, and it subsides to its normal level in December. It is of interest to notice that the peak values in both groups of diseases occur a little later in the year than the extremes of temperature.

The most striking feature in Fig. 1 is that the summer peak in mortality from intestinal diseases is gradually being reduced, but the winter peak still remains unaffected by medical methods. About fifteen years ago the chief cause of infant mortality was diarrhea and enteritis. Since 1927, the winter peak of mortality from respiratory diseases has been the most dominant feature of the mortality curve.

Environmental Factors in Relation to Respiratory Diseases

Although it is well known that the increase in morbidity and mortality from respiratory diseases follows almost exactly the occurrence of cold weather, little is known of the specific causative factors, except when the death occurs as a terminal condition of one of the contagious diseases, such as measles or whooping cough. The prevailing belief is that the increase in cold weather in the incidence of both respiratory and contagious diseases is associated with overcrowding of buildings, close personal contacts, lack of ventilation, and overheated rooms. Certain special circumstances, such as chilling and fatigue, are considered to be contributory factors, but the experimental evidence on this point is somewhat contradictory.

North (*1*)[3] ascribes the prevalence of respiratory diseases in winter partly to the lessened resistance of the mucous membrane of the nose and throat which results from the vasomotor shifts of blood to the internal organs upon exposure to cold and to changes in temperature. Others suggest that the cold air in itself may cool the blood to such an extent as to diminish the bactericidal activity of the phagocytes and the vitality of the cells which line the respiratory tract.

It is questionable whether these influences would significantly affect infants under one year of age, since these infants are usually protected from cold weather. However, a certain amount of the cooling effect may result from the rapid evaporation of moisture from the skin and mucous membranes owing to the excessively dry indoor atmospheres prevailing in winter. The usual changes in the indoor temperature may also have some effect on these infants. The deprivation of moisture from the mucous membranes may be another factor in the situation, as some believe. Recent evidence (*2*) indicates that the incidence of respiratory diseases among premature infants can be reduced considerably by proper control of temperature, humidity, and ventilation rate, and by control of contact infections.

A popular belief is that in winter we lose a great portion of the ultra violet rays of the sun, partly because of its southern declination and partly by filtration through the dusty and smoky atmosphere. Several attempts, however, to control the seasonal outbreak of respiratory diseases among children by the use of artificial ultra violet radiation proved ineffectual (*3, 4*). Abramson and Barenberg (*5*) secured much better results by control of contact infections among children than by ultra violet radiation.

The majority of the foregoing beliefs are based largely upon impressions and they are held tentatively. The problem is a highly complex one, and little exact knowledge is available on the etiology and treatment of respiratory diseases. This explains the persisting high mortality from these diseases in spite of intensive public health efforts and laboratory investigations. The most fruitful field of research seems to lie in clinical and

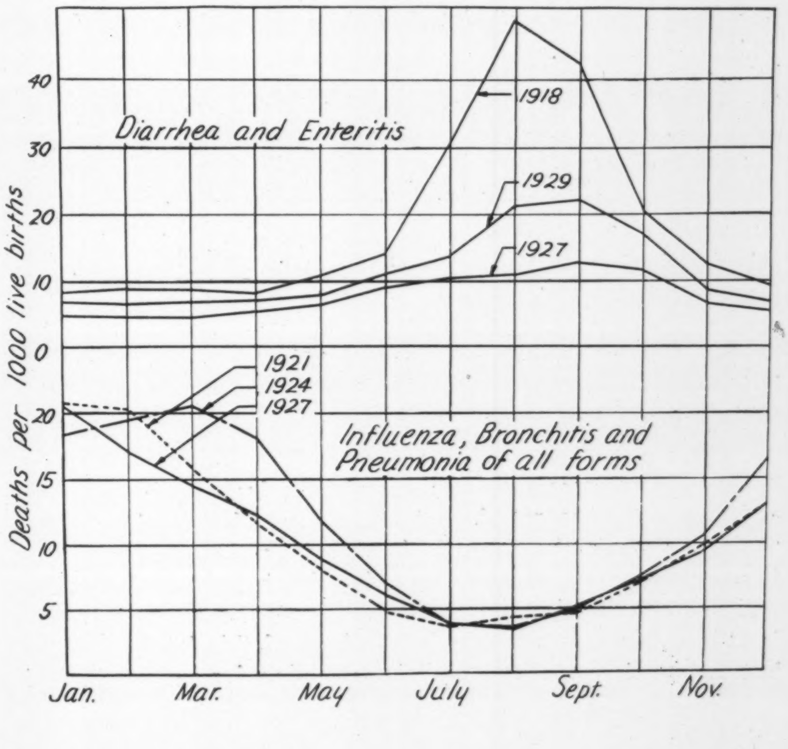

FIG. 1—SEASONAL VARIATION IN INFANTILE MORTALITY FROM RESPIRATORY AND INTESTINAL DISEASES IN BIRTH REGISTRATION AREA OF CONTINENTAL UNITED STATES

pathological studies, dealing with the entire group of respiratory diseases including the common cold. Current epidemiological studies by Frost and Doull in Baltimore, by Rosenau in Massachusetts, and by Smillie in well chosen isolated communities, will probably throw considerable light upon this obscure problem.

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[3] See bibliography to be published with the last part of this report.

Environmental Factors in Relation to Intestinal Diseases

The influence of heat upon food and other disease-transmission agents is now well recognized, but the direct effects of summer heat upon the organism are not fully appreciated. Evidence shows that high environmental temperatures, especially when they are accompanied with high humidities, cause a rise in the internal temperature, and this induces vasomotor shifts of blood to the peripheral vessels leaving the intestinal wall in an anaemic condition.

Salle (*6*) reports that exposure to temperatures of 84 to 86 F diminishes the amount of gastric secretions as well as their acidity and digestive activity. The stomach loses some of its power to act upon the food, and there is a corresponding loss in the antiseptic and antifermentive action of the gastric juice, which favors the growth of bacteria in the intestinal tract. Ample confirmation of this we find in the works of Arnold (*7, 8, 9*) and Medowikow (*10*). The latter reports, in addition, that in nine cases out of ten, *B coli* was also present in the spleen and liver of young rabbits after an exposure of twelve hours to incubator temperature.

The peculiar susceptibility of young infants to diseases of the gastrointestinal tract is attributed to their inability to adapt themselves to the climatic environment. According to Arnold (*11*), although the disturbances due to lack of adaptation can partly be offset by altering the diet, the factors of weather and foods cannot be separated, and one can influence the other.

The deleterious action of heat is, of course, a function of the indoor temperature to which the infants are exposed, but in summer this temperature depends to a considerable extent upon the outdoor temperature. Numerous studies of housing conditions in relation to infant mortality all agree that the factor leading to high indoor temperatures, such as narrow streets, overcrowding, and lack of free circulation of air due to inadequate building construction, bear heavily upon infant and child mortality.

Recent unpublished data by Blackfan and Yaglou show that (other factors remaining the same) the incidence of diarrhea among premature infants can be controlled by artificial control of environmental temperature, humidity, and ventilation rate. By properly conditioning the air with reference to these factors, the incidence was markedly reduced and it remained fairly stationary at all seasons of the year.

In the light of the evidence presented here, a campaign against seasonal diseases should include, in addition to the control of the food supply and general sanitation work, the control of the indoor air conditions, particularly in institutions for sick infants and children. Protection against excessive summer heat is still a rare exception in institutions of this kind, but the number of new hospitals providing for artificially cooled rooms for infants is gradually increasing.

AN OFFICE AIR CONDITIONED FOR COMFORT AND HEALTH

Geographic Distribution of Seasonal Diseases According to Seasonal Temperature

The literature on the geographic distribution of morbidity and mortality from respiratory diseases is quite contradictory. The consensus seems to indicate that there are no marked differences in the various parts of the United States.

In a very thorough study of the local and sectional characteristics which affect the infant mortality, Holland and Palmer (*12*) came to the conclusion that climatic factors, such as temperature, rainfall and sunlight, are relatively unimportant in their effect on urban infant mortality. Race, nativity, economic and educational status are claimed to be the chief factors.

Huntington (*13*), on the other hand, finds a definite relationship between temperature and daily deaths in New York City for all ages as well as for children under five years of age. In this latter age group, the deaths were found to decline progressively as the temperature rose to 55 F, but increased portentously with every rise of temperature above 60 F.

It must be conceded that the causes of infant mortality are very complex, and so closely related to community life and general environment that it is difficult to separate and evaluate the effects of the various climatic factors. Nevertheless, it would be equally difficult wholly to explain on any other grounds the fact that some cities on the Pacific Coast, which are blessed with a climate approaching the ideal for infants, now have and have had for many

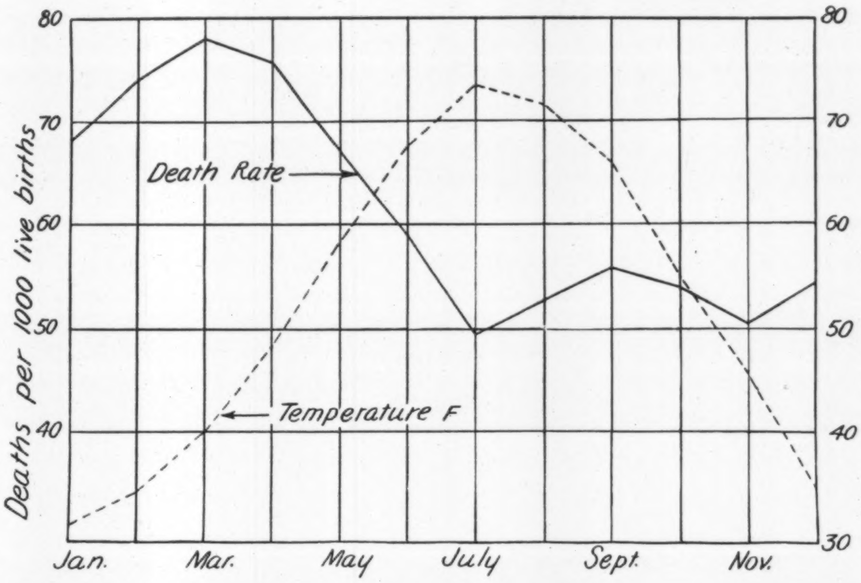


FIG. 2—SEASONAL VARIATION IN INFANTILE MORTALITY FROM ALL CAUSES IN NEW YORK CITY, 1925 TO 1929

years the lowest infant mortality in the United States. Recent public health efforts were successful in reducing infant mortality, but the decline has not altered appreciably the relative geographic distribution of mortality.

In order to test for geographic distribution of infant mortality on the basis of climate, a special study was undertaken by the committee on growth and development. Owing to the complexity of the problem, exact mathematical methods were not justified, but certain empirical methods had to be used for the purpose of eliminating and balancing, as much as possible, the unwanted factors. To begin with, the powerful influence of race was eliminated by considering white infants only. The method of analysis used is based upon the fact that, in the temperate climates of the United States, the weather at some period in the course of a year is apt to approach a more or less favorable state. Under this condition, the influence of climate would be at a minimum, and the actual mortality at this particular period, or a little later on in the year, would depend largely on other local and sectional factors which affect the infant at all times. Starting with this as a basic level of infant mortality for any given community, the increase in mortality from this basic level to the crests of the winter and summer waves must be largely due, either directly or indirectly, to the unfavorable weather condition prevailing in that locality. The seasonal excess in infant mortality computed in this manner was found to bear a fairly close relation to seasonal temperature, regardless of whether the lag between the extremes in temperature and the extremes in death rate was taken as one or two months.

As an example, Fig. 2 shows the characteristics of the seasonal trend of mortality for the city of New York. As a general rule, the minimum infant mortality in New York occurs in July and the peaks of mortality occur two months after the month of extreme temperatures. For the group of cities used in this study, the average lag between the extremes in temperature and the extremes in death rate was found to be about one month.

If now the mortality rates for the peak winter months (usually February, March and April, as in New York) are averaged, and from this average the minimum mortality (usually that in July) is subtracted, the difference gives the winter excess in mortality. This is tabulated against the average temperature for January, February, and March (one month lag). In the same manner, the minimum mortality is subtracted from the average mortality for the peak summer months (usually August, September, and October) and the difference tabulated against the average temperature for July, August, and September.

Altogether, forty-one cities were used in this analysis. The infant mortality rates for thirty-eight of these were secured from the weekly reports of the U. S. Public Health Service (1925-1929), and for the remaining three cities, namely, Birmingham, Memphis, and New Orleans, from city health departments. The thirty-eight cities represent those in the birth registration area of continental United States since 1925, or earlier, for which Weather Bureau data are available. An exception had to be made for the city of Des Moines, which was excluded from the analysis because the mortality rates varied so much that it was difficult to decide with any degree of certainty upon the maxima and minima. Unfortunately, it was impossible to secure data on more than forty-one cities, on account of limitation in time.

Concerning the years to be investigated, there was little choice. The five consecutive years, 1925 to 1929, had to be studied because the mortality data for white infants prior to 1925 are meager. The results, therefore, include the effects of the epidemics of influenza and pneumonia which occurred in the years 1926 and 1928-1929.

In deriving the general trend, the cities were first classified according to their seasonal temperature, as in Table 1. The weighted average seasonal excess in mortality was then computed for each temperature group by multiplying the seasonal excess of mortality in each city by the corresponding number of births for the year 1927 (1928 and 1929 birth statistics not yet available) and dividing the sum of the products by the sum of the births. In the same manner, the weighted average seasonal temperature was computed. Table 1 shows the results.

The following three main features are to be noted in this table:

(a) Minimum infant mortality appears to occur when the sea-

Table 1—Geographic Distribution of Seasonal Excess in Infant Mortality According to Seasonal Temperature

Temperature Groups	Number of Cities in Each Group	Weighted Average Seasonal Temperature		Weighted Average Seasonal Excess in Infant Mortality Rate	
		Winter	Summer	Winter	Summer
F		F	F		
10 —14.9	1	12.4	....	15.1	....
15 —19.9	2	17.8	....	22.6	....
20 —24.9	0	....	....	....	....
25 —29.9	7	27.2	....	30.3	....
30 —34.9	15	32.2	....	26.3	....
35 —39.9	6	35.5	....	26.3	....
40 —44.9	4	42.2	....	13.0	....
45 —49.9	2	46.7	....	9.9	....
50 —54.9	1	52.2	....	21.0	....
55 —59.9	4	58.3	....	12.9	....
60 —64.9	5	....	62.6	....	3.2
65 —67.4	6	....	67.8	....	7.0
67.5—69.9	12	....	69.0	....	9.6
70 —72.4	7	....	70.3	....	7.0
72.5—74.9	6	....	73.1	....	18.8
75 —77.4	4	....	75.4	....	23.6

sonal temperature, as computed in this study, averages between 60 and 65 F.

(b) The mortality increases for temperatures above and below this apparent optimal range.

(c) There is an indication for very low temperatures to be associated with a relatively low mortality.

On the whole, the general trend is fairly clear, but there is some distortion for temperatures between 50 and 60 F which may be due to the small number of cities involved. The chief cause, however, is found to lie in the three California cities used in this study, San Francisco, Los Angeles, and San Diego, all of which for some unknown reason have excessively high mortality rates from respiratory diseases for their climate.

Fig. 3 shows the crude trend, excluding the three California cities, and the relative position of these cities with reference to the general trend. With one or two exceptions, the general characteristics of Fig. 3 are fairly in accord with those found by Huntington (*13*) for children under five years of age, in the city of New York. The relatively low death rate for temperatures below 25 F has been observed by Huntington also, but his curves do not show an actual drop in mortality as does Fig. 3. Huntington attributes the flattening of his curves chiefly to the small number of observations, an explanation which may also hold true in the present study. It is possible, however, to explain this by the effects of extreme cold upon the parasitic cause of disease or upon the human organism itself.

Although infants under one year of age are, as a general rule, protected from cold weather, Fig. 3 shows a progressive decrease in mortality with progressively increasing seasonal temperatures, from 30 to 60 F. This is an unmistakable characteristic of the respiratory group of diseases, for which temperature appears to be the best known index of the effects of changing seasons.

In summer, as the seasonal temperature of the localities increases above 65 F, the mortality, chiefly from diarrhea and enteritis, appears to increase sharply almost in direct proportion to the temperature. An exception to this is to be found in the excessively warm cities of the South where the mortality, after reaching peak values early in May and June, begins to decline while the temperature is still high, or even rising. In the light of Arnold's studies (*11*) these facts may be explained partly by the adaptive changes which take place in the defensive mechanism of the intestinal mucosa, after a period of exposure to high temperatures, and partly by changes in diet.

According to Fig. 3, not only the effects of cold but also the effects of heat upon infants, whether the direct, or indirect, effects, or both, are still at large and far from being under adequate control. This holds true particularly in cities in which little public health work is being carried on for the control of the intestinal group of diseases. As a general rule, these cities lie above the general trend shown in Fig. 3.

In spite of the apparent definite relationship between seasonal temperature and seasonal variation in infant mortality, it should be made clear that we are relying on meager data and upon massed aver-

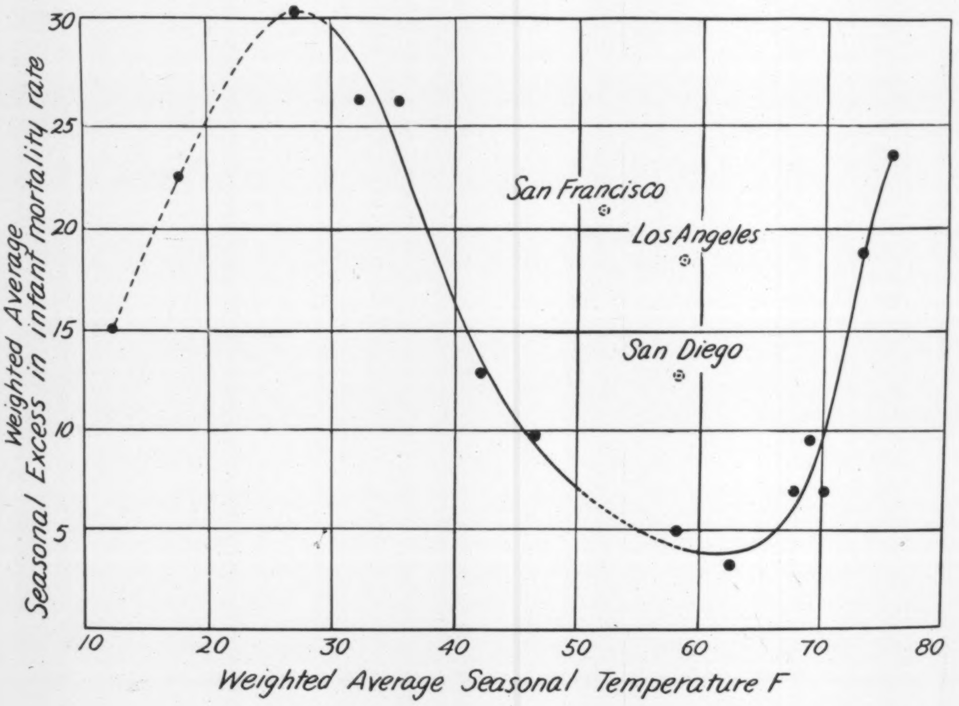


Fig. 3—Geographic Distribution of Seasonal Excess in Infantile Mortality Rate According to Seasonal Temperature in a Group of Forty-One Large Cities in the United States, 1925 to 1929

ages in which the effects of many other factors, directly or indirectly associated with temperature, are imperfectly separated from those of temperature. For this reason Fig. 3 may be of little value in so far as quantitative data are concerned. Its value lies chiefly in showing the prevailing crude trend of climatic influence, for which temperature appears to be the most important index.

The recent introduction of several more states to the United States birth registration area may yield sufficient data to separate the effects of some important factors which affect the infant mortality, and derive a revised trend which would be more truly representative.

The Properly Conditioned Air Supplied This Textile Mill Is Essential to the Processing of Materials and Also Provides Comfort for the Employes

The Influence of Humidity, Variability, Wind, and Other Factors, Upon Infant Mortality

According to Huntington (*13*), atmospheric humidity plays only a minor rule in children under five years of age as compared with older persons. He believes that if it were possible to elminate the effects of all other climatic factors, one should find that at all temperatures up to the optimum (55 F according to his data), the best relative humidity exceeds 80%.

The present study discloses a significant humidity effect when individual cities are considered by themselves. In Table 2, the monthly infant mortality rates for New York City are classified in two groups according to the average monthly relative humidity. Although the range from high to the low humidity groups is quite small, it can be seen that for temperatures up to our probable optimum of 63 F (Fig. 3), the higher humidities are, as a general rule, associated with the lower infant mortality. To some extent, this confirms Huntington's finding to the effect that a moist atmosphere is more favorable for respiratory diseases than a dry one, but it is diametrically opposite to what some persons held and taught for many years. In July and August. when the temperature exceeds 65 F, the reverse holds true, as in Huntington's data. These general tendencies for New York City are more or less representative of several other large cities studied by the committee. Unfortunately, the number of cities we are dealing with and the range in the seasonal humidity between the various cities, are too limited for the purpose of separating the effects of humidity from those of temperature in the general trend shown in Fig. 3.

In addition to these statistical data, accurately controlled studies on premature infants (*2*) also show that a fairly high degree of moisture in the air is associated with a lower mortality among these infants.

Variability in temperature and wind did not appear to exert any significant influence upon infant mortality, when the data for large cities were analyzed by a method similar to that used for humidity (Table 2). For persons over five years of age, however, Huntington (*13*) states that a moderate high degree of variability in temperature from day to day and variable winds of moderate intensity, are stimulating and are associated with a lower death rate.

In the light of our study, precipitation appears to be a factor of some importance, but the influence of other elements, such as sunshine, cloudiness, storms, etc., render the problem too complicated to be considered here.

The Probable Optimum Climate

Both statistical and experimental studies indicate that there can be no ideal climate to suit every purpose. The state of health, state of civilization, race, type of disease, medical methods, age, etc., are all factors which may alter the optimum of climate to a considerable extent.

For inhabitants of New York City over five years of age, Huntington's optimum climate consists of a temperature of 65 F, a relative humidity of nearly 90%, and a moderately high degree of variability from day to day (*13*).

For children under five years of age, which means mainly for infants under one year of age, Huntington tentatively sets the optimum temperature at 55 F and the humidity at 80% (*13*).

According to the present study the probable optimum temperature for infants under one year of age appears to be about 63 F (Fig. 3) or about 8 F higher than Huntington's value. If this is true, the optimum for

Table 2—Effect of Humidity upon Infant Mortality Data for New York City for the Years 1925-1929

Monthly Groups	Number of Months in		Average Temp. F	Range in Relative Humidity %	Average Relative Humidity %	Average Infant Mortality Rate
	High Humidity Group	Low Humidity Group				
Jan. &	5	....	32.8	67–69	66.8	68.5
Feb.	....	5	32.8	58–64	62.2	73.4
March &	4	....	45.4	62–68	64.7	69.7
April	....	6	44.3	57–59	57.6	81.5
May &	4	....	63.8	67–70	67.8	58.2
June	....	6	63.2	56–63	60.8	65.2
July &	3	....	71.6	73–75	74.1	52.8
Aug.	....	7	73.0	67–69	67.7	50.2
Sept. &	4	....	63.6	71–73	71.7	55.5
Oct.	....	6	58.6	66–69	67.9	54.5
Nov. &	3	....	43.6	69–74	73.6	49.6
Dec.	....	7	39.1	63–67	65.5	53.9

these infants is substantially the same as that of Huntington's for persons over five years of age.

The apparent difference between Huntington's and the writer's optimum for infants may be due to the fact that his data deal with deaths in 1882 to 1888, whereas the present data include deaths in 1925 to 1929. The recent significant reduction in infant mortality from diarrhea and enteritis by control of the parasitic cause of the disease may be responsible for the apparent increase in the optimum climatic temperature. This stands to reason if one considers that the optimum temperature, as determined statistically, is mainly a compromise between the physiological optimum for the human organism itself, the optimum for the respiratory group of diseases, and the optimum for the diarrheal group. Further advances in medical methods and improvements in general sanitation work may alter the optimum from year to year until it may finally approach the physiological optimum for the human organism itself.

We must now distinguish between indoor and outdoor atmospheres. The infants we are dealing with are not, of course, exposed to the rigors of cold but they are well protected in artificially heated rooms. Nevertheless, they are affected by adverse weather because the outdoor climate modifies the indoor atmosphere and the mode of life. Temperature by itself may not be the primary causative factor, but it is the most conspicuous characteristic of the changing seasons, and the best index we have, at present, of climatic effects upon the organism. Overcrowding and lack of ventilation accompany cold weather. An optimum climatic temperature of 60 to 65 F may in reality mean that the weather is mild and pleasant enough to take the infants out-of-doors or to secure better ventilation indoors by opening windows.

The solution of the general problem of climate and health requires extensive studies of the effects of many other elements, such as atmospheric ionization, sunshine, barometric pressure, precipitation, wind, frost, and perhaps other factors, all of which may have an important bearing upon health.

Seasonal Variation in the Growth of Children

Statistical studies reveal a seasonal periodicity of growth in children. Porter (*14*) has shown, from measurement of thousands of Boston school children, that the months from July to December are especially favorable for growth in weight. The most rapid gain was found to occur from September to January, the average gain during the period being four times that during February to June.

In Johannesburg, Transvaal, where the seasons are reversed, Cluver (*15*) found a corresponding reversal in the period of rapid growth in weight of infants. The infants gained nearly twice as rapidly in the cool season, April to September, than in the warm season, October to March.

Nylin's observations (*16*) are of special interest. His evidence indicates that school children in Spanga and Stockholm grow fastest in height in spring, but gain in weight most rapidly in late autumn and early winter. On the whole, the weight increase seems to vary inversely with the height increase. Nylin does not agree with the prevailing view that the seasonal variations in growth are due to variations in the water content of the body. Although he appreciates the influence of sunlight upon growth he points out that his findings do not confirm this connection and that further studies into unidentified endogenous factors are needed in order to clear the situation.

Atmospheric Pollution

The most important atmospheric pollution at present is that due to smoke and soot discharged from household and industrial chimneys in many large cities. In addition to the suspended carbon, dust, and ash particles, the exhalations are charged with unconsumed hydrocarbon gases, sulphur dioxide gas, and other industrial fumes capable of injuring property, vegetation, and health. Aside from specific industrial processes which produce harmful dusts and fumes, the consensus of opinion is that most of the mischief attributed to general atmospheric pollution is fictitious and generally indirect. The objection is considered to be largely economic.

Dust may affect health in two ways; (a) by inhalation and (b) by obliteration of the sun's rays. Although these facts are well recognized, there is no proof that the effect of dust in ordinary air is sufficient to cause demonstrable injury to health. The effects may be slow, cumulative, and insidious in their manifestations, but for the time being direct evidence is lacking.

Dust-borne bacteria are sometimes held responsible for the transmission of disease microbes through the medium of air, but the epidemiological evidence weighs on the negative side. Bacteria of the pathogenic variety were found to constitute but a very small proportion of the total dust-borne bacteria (*17*). The non-pathogenic bacteria in air are, in general, considered harmless, but it is not known whether the constant inhalation of these organisms may eventually produce harmful effects.

Carbon Monoxide

Another prevalent contamination of air is automobile exhaust gases. The most important of these is carbon monoxide. During heavy traffic hours, the air of congested streets and boulevards is found to contain enough carbon monoxide to menace the health of those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. Under ordinary conditions, however, the concentration of carbon monoxide in city air is believed to be insufficient to affect the average city dweller or pedestrian. Stevens (*18*), on the other hand, finds that infants and young children are peculiarly susceptible to small but repeated amounts of carbon monoxide gas in the air they breathe. His studies seem to disprove the long accepted theory that carbon monoxide is not a cumulative poison, and they suggest the desirability of guarding infants and young children from unnecessary exposure to such atmospheres, either in the home or in congested city streets. As far as the domestic hazard from carbon monoxide is concerned, the flueless gas heating stove is mainly responsible (*19*). Gas heaters should be provided with flues, to carry off all combustion products.

In the light of our meager knowledge of the effects of atmospheric pollution on health, it is fair to say that any conspicuous air pollution constitutes a liability to property and health, and that steps should be taken to reduce the pollution to a minimum. This would require the concerted efforts of the fuel engineer, the public health officer, and the public itself.

(*To be concluded in the next issue*)

Piping
High, Intermediate and Low Pressure Steam Economically

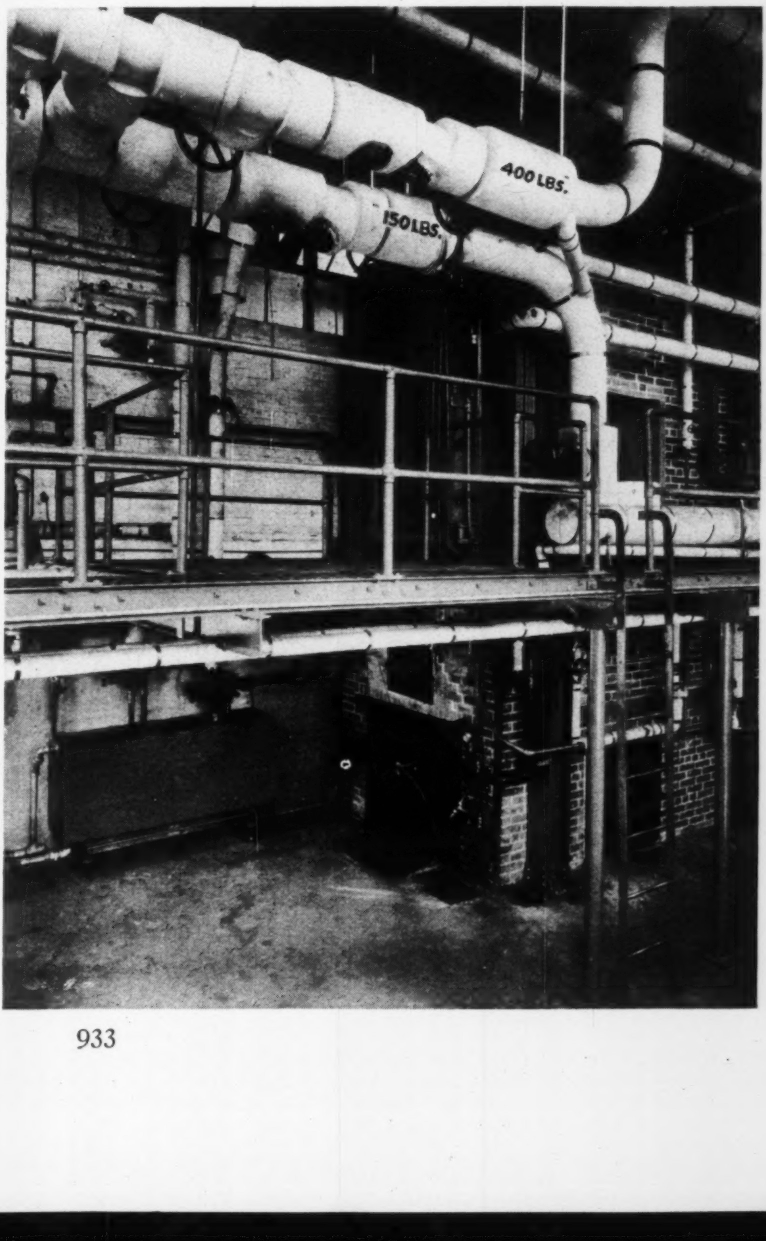
In addition to piping heating and process steam, the system described in this article supplies high pressure steam for the testing and development of various pressure regulating devices. The arrangement of equipment and piping for accomplishing these purposes effectively and economically is discussed and illustrated

By H. L. Colby*
and
Francis A. Westbrook†

THE NEW boiler plant recently installed at the plant of the Mason Regulator Company in Boston is of peculiar interest because it has been ingeniously designed for a double purpose. In addition to making provision for heating and process steam, it has been necessary to supply high pressure steam for the testing and development of various pressure regulating devices. With this installation, steam at a normal pressure of 150 lb. is supplied to all lines centering in the boiler house, except that when desired 400-lb. steam can be delivered to the high pressure testing laboratory.

Arrangement of Reducing Valves

How this is accomplished is shown in Fig. 1, which shows, schematically, the layout of piping and valves. Thus, at normal 150-lb. operation, valves *A* and *A′* are opened and valves *B* and *B′* are closed. Under these



THIS ILLUSTRATION SHOWS THE 400-LB. LINE, WHICH CONVEYS STEAM FOR THE HIGH PRESSURE TESTING LABORATORY, THE 150-LB. LINE, AND THE OIL BURNER conditions 150-lb. steam goes into all parts of the piping system throughout the plant for testing, processing, heating, feed water heating and preheating of the oil used for the oil burner under the boiler. Process steam lines are taken from the 150-lb. pressure mains through reducing valves introduced at points where steam at lower pressures is required. Unit heaters are supplied with steam through one reducing valve serving the whole group. The feed water heater and oil preheater are served through reducing valves in parallel at 150 lb. to 30 lb. and the office heating line has reducing valves at the office building for providing pressure suitable for the radiators.

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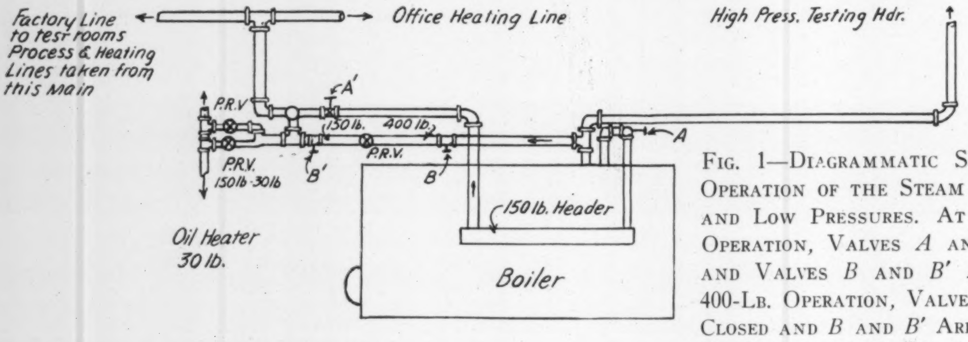


Fig. 1—Diagrammatic Sketch to Show Operation of the Steam Piping at High and Low Pressures. At Normal 150-Lb. Operation, Valves *A* and *A'* Are Open and Valves *B* and *B'* Are Closed. At 400-Lb. Operation, Valves *A* and *A'* Are Closed and *B* and *B'* Are Open So That 400-Lb. Pressure Is Supplied the Line to the Laboratory, But Is Reduced to 150-Lb. for the Rest of the System

At times when the high pressure testing laboratory requires 400-lb. steam, valves *A* and *A'* are closed and *B* and *B'* are opened. This permits the high pressure steam to flow direct to the laboratory but, by means of a 400-lb. to 150-lb. reducing valve placed between *B* and *B'*, 150-lb. steam, as before, is supplied to the remainder of the system.

Equipment Used

With this lay-out and its purposes in mind it will be interesting to see what equipment has been used. A 93-hp. sectional type water tube boiler suitable for 400-lb. working pressure and arranged for oil burners has been used. The trimmings and accessories consist of:

1 water column with high and low water alarm, gage and gage cocks, operated by chains from the boiler room floor.
1—12 in. steam gage.
2—1½ in. pop safety valves.
1—2 in. extra heavy, heavy feed flanged stop valve.
1—2 in. extra heavy check valve.
1—2½ in. extra heavy blow-off valve.
1—4 in. tube cleaner with a metal hose to operate on 75 lb. city water
3 side dusting doors with frames installed in the settings.
3 cleaning doors with frames installed in the settings.
1 brick lined furnace door for the setting, as well as baffle tiles to provide for three passes of the hot gases of combustion over the boiler tubes, together with the steel supporting columns and stays for the boiler.

Boiler Feed and Accessories

The boiler feed pump consists of a 6 in. × 3 in. × 6 in. duplex bronze fitted pump with a maximum discharge pressure in excess of 400 lb. and a capacity of 14 g.p.m. It is equipped with a pump governor and water feed valves. In addition to the feed pump, there are two inspirators for emergency service. One of these is for use with the 400-lb. working pressure and has a capacity of 1250 gallons per hour. The other is to be used for the 150-lb. working pressure. A suitable suction tank has been provided for these inspirators and it has been equipped with a 1¼ in. cold water feeder with float and lever attachment in order to maintain, automatically, the proper water level.

The return tank is furnished with a 1-in. balanced valve to serve as the water feeder. It is operated by an automatic ball and float attachment in the tank which maintains the predetermined water level.

The feed water heater is a standard 4-tube heater containing 50 sq. ft. of copper tube heating surface, the tubes and water section being suitable for 400-lb. working pressure.

By-pass for Feed Water Heater

A by-pass connection has been provided for the feed water heater and inspirators (see Fig. 3) which has been so valved that the pump can be made to discharge either through or around the heater. The inspirators can deliver the feed water direct from the tank served by city water. As previously stated, the inspirators have valves whereby it is possible to use either the one for 400-lb. steam pressure or that for 150-lb. independently. Two 1-in. relief valves have been installed, one on the feed water heater and one on the discharge from the 150-lb. inspirator which are set for 400 lb. and 175 lb., respectively, to provide protection.

The Oil Burner and Its Control

An oil burner especially adapted to variable load conditions has been selected. It is controlled automatically by a diaphragm-operated mercury switch, which is actuated by drop in boiler pressure below the normal of 150 lb., for which it is set. This switch, in turn, governs the operation of a control motor on the oil feed valves.

A 5000-gal. oil tank is buried in the yard outside of the building. There is an electric immersion-type preheater and a steam preheater for the oil. When starting up the cold boiler the oil is electrically preheated, but when the steam reaches the proper pressure the steam preheater is cut in and the electrical preheater cut out.

In starting up the cold plant it is necessary first of all to open the main stack damper, which at the same time automatically operates a switch to cut in the electrical current. The electric oil preheater is then thrown on by hand through a tumbler switch until the oil reaches

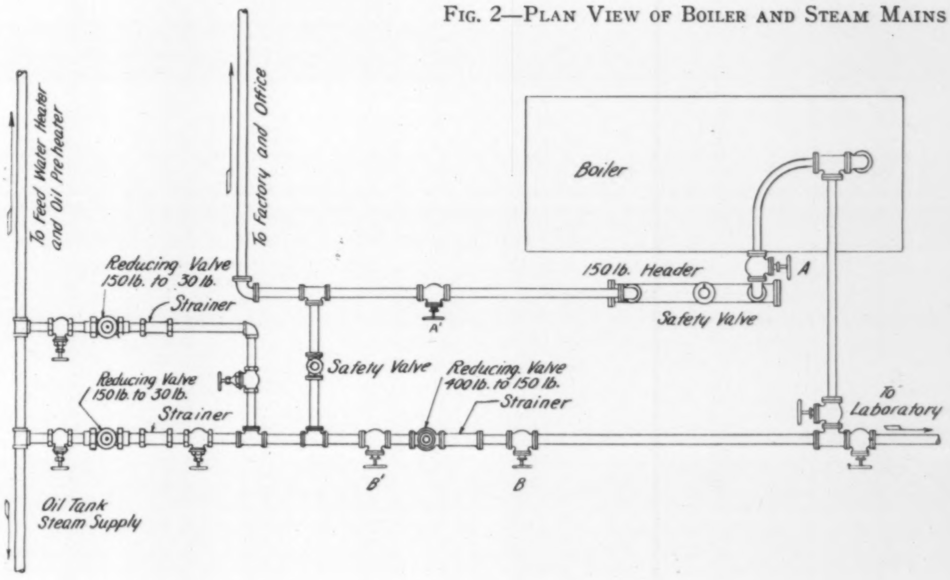

To Feed Water Heater and Oil Pre heater
To Factory and Office
Boiler
150 lb. Header
A
Safety Valve
A'
Reducing Valve 150 lb. to 30 lb.
Strainer
Safety Valve
Reducing Valve 400 lb. to 150 lb.
Strainer
Reducing Valve 150 lb. to 30 lb.
Strainer
To Laboratory
B'
B
Oil Tank Steam Supply

FIG. 2—PLAN VIEW OF BOILER AND STEAM MAINS

a temperature of 110 F, after which the oil pumps are started from an electrical push button station.

The burner is then started and with the pipe lines closed off to the system, pressure is permitted to build up in the boiler to 125 lb. At that time steam is turned on to the system, which includes the steam oil heater. The electrical heater is then manually cut out by the tumbler switch on that unit.

Oil is supplied from the tank to the two burners by means of two heavy duty oil pumps direct connected to 1 hp., 3 phase, 60 cycle, 220 volt induction motors.

Piping Materials

The piping for an installation being an important part of the job as much from the standpoint of expense as from operation, it will be of interest to see what materials have been used. For cold water, piping is standard galvanized with cast iron screwed fittings and brass gate and check valves. Brass pipe is used for the inspirator suction lines and the overflow from the inspirator tank.

The inspirator and boiler feed pump discharge lines are designed for working pressures in excess of 400 lb. Consequently extra heavy steel pipe, cast steel fittings, and forged and cast steel globe and check valves suitable for such pressure have been used. Valves 2-in. and larger are flanged with outside screw and yoke; and on smaller lines the valves have forged steel flanges or steel unions with monel metal seats and ground joints.

Wrought iron pipe is used for exhaust steam from the boiler feed pump, the vapor pipe from the return tank, the vents from the return mains and the safety valve and boiler blow-off piping. The safety valve piping has extra heavy screwed fittings. The two 1½-in. pop valves on the boiler and the 2½-in. safety valve on the 150-lb. steam header are piped to a 4-in. blow-off line with provision for a future boiler. The blow-off piping is of 2½-in. extra heavy wrought iron with extra heavy cast iron screwed fittings. A 400-lb. blow-off valve has been installed. The blow-off line also has a 2½-in. outlet for a future boiler.

All of the piping connected directly to the boiler and carrying boiler pressure must of course be suitable for the higher, or 400-lb. pressure, used. For this reason it is of extra heavy steel pipe with welded steel joints wherever possible and otherwise with forged steel fittings and flanges back welded to the pipe ends.

The angle and check valves in these lines are of forged steel construction depending on their size, location and service. Two of the main valves have chain and sprocket control from the floor. Details of the high pressure steam piping are shown in Figs. 1 and 2.

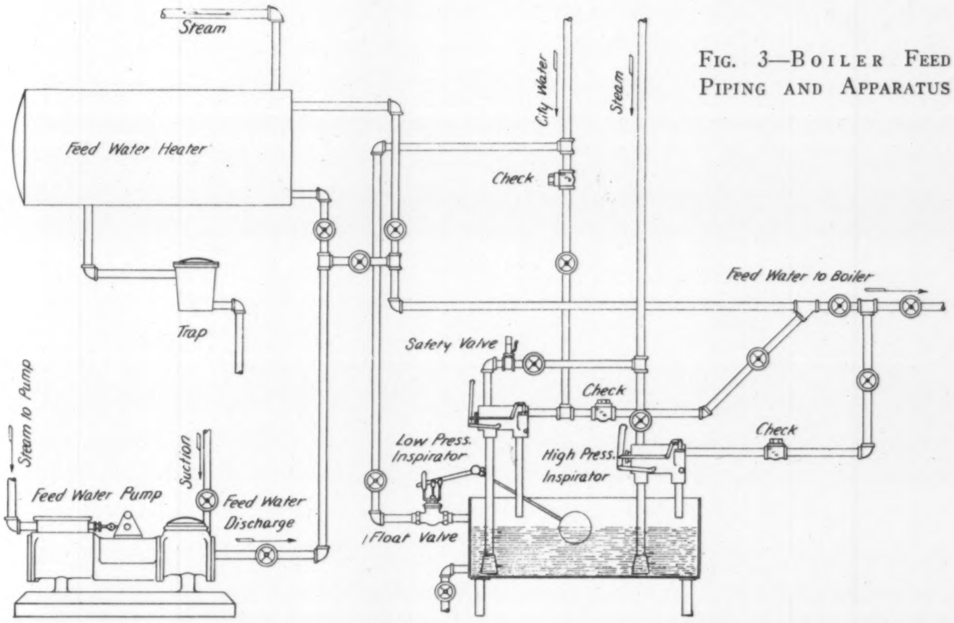
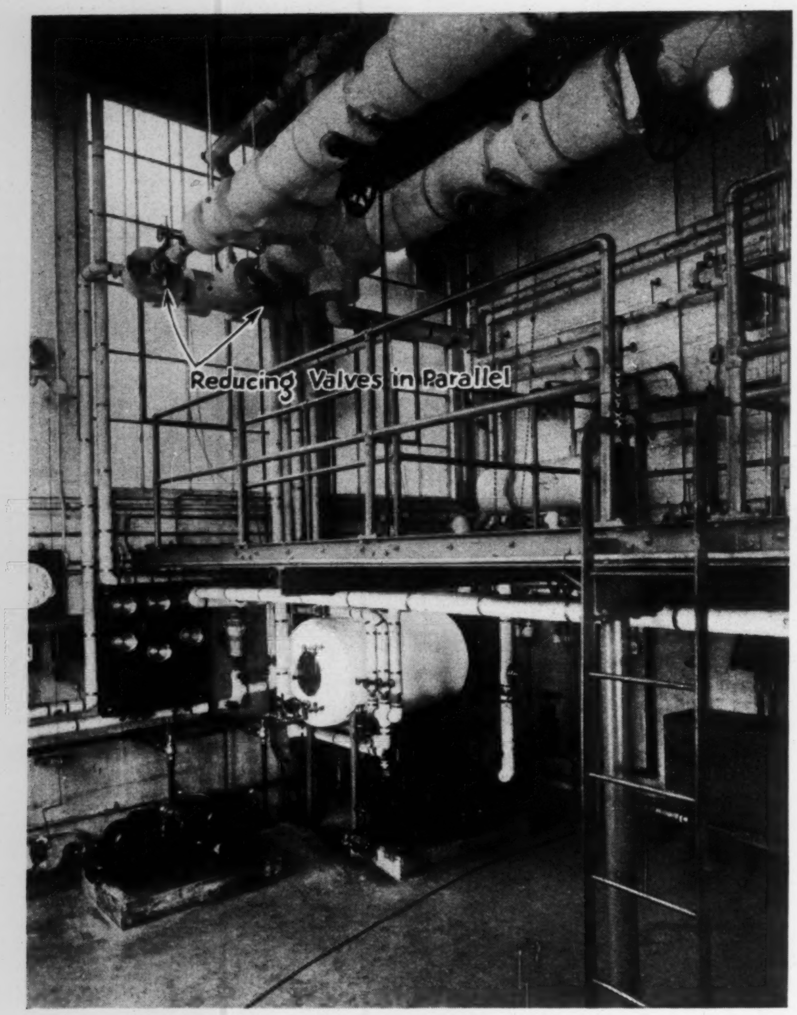

Steam
City Water
Steam
Feed Water Heater
Check
Trap
Feed Water to Boiler
Safety Valve
Check
Check
Steam to Pump
Suction
Low Press. Inspirator
High Press. Inspirator
Feed Water Pump
Feed Water Discharge
Float Valve

FIG. 3—BOILER FEED PIPING AND APPARATUS



An interesting feature is the parallel arrangement of two reducing valves of different sizes in the main 150-lb. steam line serving the whole plant. The smaller is for the summer load when the unit heaters are not in operation and the larger for the heavy winter load. The large valve is set for a slightly lower pressure on the low pressure side so that when the demand for steam increases beyond the capacity of the small reducing valve it opens automatically.

Parallel Arrangement of Reducing Valves, One for the Summer Load When the Unit Heaters Are Not in Operation, and One for the Heavy Winter Load

THREE OF THE ARTICLES TO APPEAR IN THE DECEMBER ISSUE

D. D. Kimball will describe the development of a college power plant in next month's HEATING, PIPING AND AIR CONDITIONING. How a surplus of exhaust steam was utilized in order to prevent wastage and excessive fuel consumption is told in detail and the growth of the plant which serves Berea college and which was given a rating of the highest class for installation and operation by engineers of the U. S. Fuel Administration in 1918 is discussed.

In a special press room of a printing plant the air conditioning equipment is used intermittently at full capacity in order to effect economies. In his paper to appear in the December number, Charles S. Leopold points out the reasons for the intermittent operation of air conditioning systems, discusses their proper control and presents test data and results for this typical industrial installation.

Oil in exhaust steam used for heating or process forms a film on heat transfer surfaces and cuts down the amount of heat transmitted materially. In order to insure the efficiency of such equipment it is necessary that the prime mover and auxiliaries be lubricated by a correct oil applied in the proper manner. Next month, R. P. Tobin's article tells the "why" and "how."

—and numerous others. The index for 1931 will also be published.

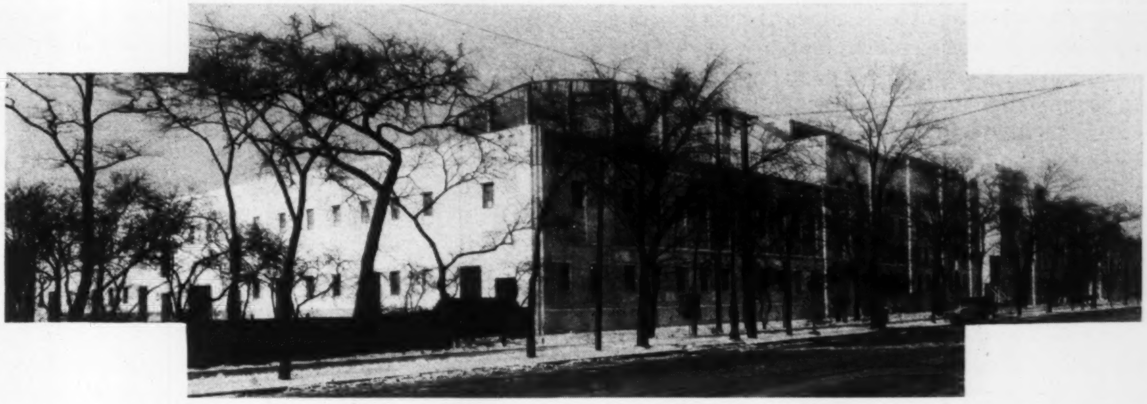


FIG. 1—THE ARMORY OF THE 124TH FIELD ARTILLERY IN CHICAGO, WHICH IS HEATED BY GAS-FIRED UNITS. THE DRILL HALL AREA IS LARGE ENOUGH TO PERMIT A BATTALION OF CAVALRY TO MANEUVER

Eight Gas-Fired Units Heat Large Armory

By
Louis L. Narowetz Jr.*
and R. M. Moffitt†

WHAT is said to be the largest gas-fired unit heating job in the world is that in the armory of the 124th field artillery in Chicago, shown in Fig. 1. This structure—outstanding in military as well as in architectural circles—is of steel and stone construction. The drill hall is 224 by 336 ft. and has a cubical content of 5,750,000 cu. ft. More than half of the drill hall area is used for the arena, in which the maneuvering of a complete cavalry batallion is possible; 5,000 spectators can be seated in balconies on the east and west sides of the drill arena.

There are two glass sky-lights, each 36 × 330 ft. on each side of the center of the building. Along each side of the balcony are windows with a total glass area of 864 sq. ft. There are rooms under the balcony which are heated by gas steam radiators, but heating for the comfort of the spectators in the balcony is the part that required the greater amount of study on the part of the engineers and contractors.

Eight Unit Heaters Used

For this purpose, eight gas-fired unit heaters of the cast iron sectional type are used. Each of the four end units has an output capacity of 800,000 Btu. The intermediate units deliver 700,000 Btu each. Each unit is enclosed in an insulated casing and arranged for downward air travel through the heater; the flue gas travel within the heater sections is upward, promoting efficiency.

Air delivery is effected by a blower for each unit, delivering 14,500 and 12,700 c.f.m., respectively, to the end and intermediate units. The blowers are driven by direct connected motors, each being controlled from a switch panel on the balcony floor.

The fuel control is so arranged that the gas is extinguished when the air temperature within the heater casing reaches 110 F. This prevents over heating damage due to the operation of the heater without the blower, and of special importance, prevents abnormal outlet temperatures where excessive bouyancy is undesirable.

The units are fixed in the truss space in the alternate bays 64 ft. apart and 46 ft. above the arena floor as shown in Fig. 2. A 6-in. gas main runs on each side of the balcony in the truss area with a 2-in. branch to each unit. Each blower is mounted on the same structure with its heating unit.

Recirculation Ducts Installed

To provide adequate distribution of air, recirculation ducts were installed drawing air from sixteen grilles in the face of the arena walls. These are 48 in. long by

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† L. J. Mueller Furnace Company, Chicago, Ill.

18 in. deep and are located 10 ft. above the floor and 16 ft. apart. The recirculated air is brought back to the fan, forced through the gas-fired unit heaters, each of which discharges the heated air through three supply ducts arranged with volume and deflecting dampers so as to deliver the heated air directly over the space occupied by the spectators, as shown in Fig. 3.

Products of Combustion

The products of combustion are carried away through copper molybdenum alloy flues. Since no draft tension is required or desired for gas fuel, high stacks are not necessary. In this case the flue gases are released at but a few feet above the roof; the vents are not visible from the street. The connection of each heater section, having its own individual burner, pilot and vent, to the flue header, is through individual 4-in. back draft diverters. The header is pitched to drain condensed flue products which are in turn diverted to the sewer.

Calculating the Requirements

Little data were available with reference to a gas heating job of this size and character; for this reason it is believed that the manner in which the calculations were made and the results secured will be of interest to engineers generally. In computing the heat requirements for those sections of the building to be heated, no direct recognition was given to the vast space over the drill arena proper, nor to the roof exposure over this space. To the computed heat loss of the balcony areas, 20 per cent was added arbitrarily for diffusion losses to the arena space. Subsequent performance has shown this to be more than the actual excess requirements.

Floor space for heaters or blowers was not available under any circumstances. The problem resolved itself into removing air from the bottom of the balconies in sufficient quantities to more than neutralize the buoyancy of the heated air above. Low outlet temperatures and correspondingly large volumes are used. From smoke tests it has been found that diffusion is effected evenly. The real work of temperature distribution is done by the recirculating grilles located just beneath the lowest balcony level, *ten feet above* the arena floor line. With a 65 F required balcony temperature and a 35 F rise through the heaters, the outlet temperature is 100 F. With definitely known unit capacities, the correct air volumes were determined.

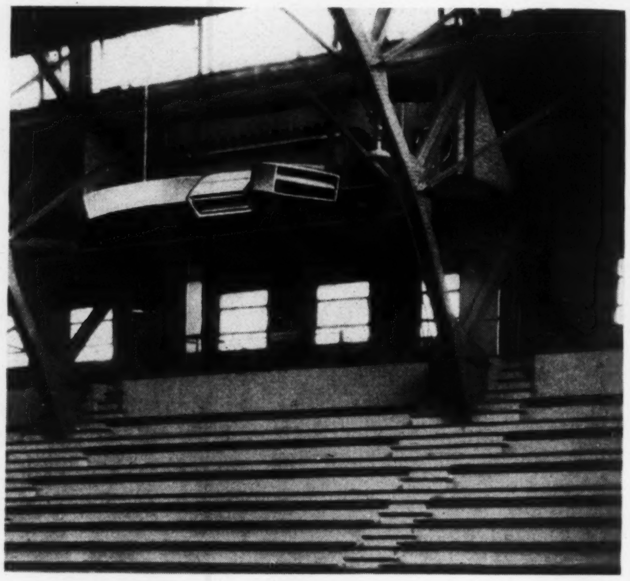

FIG. 3—EACH UNIT DISCHARGES THE HEATED AIR THROUGH THREE SUPPLY DUCTS

The plant has operated through one heating season very satisfactorily. It possesses flexibility for intermittent heat. Fifteen minutes after the valve is turned any or all the units are operating at capacity. Fuel is used only for that part of the balcony to be heated and only for the period requiring heat.

Operating cost has been found to be low and is comparable with coal fired steam. Structural economies were also effected.

Confining heated air within limits in unconfined spaces and, as in the case of the roof and the extraordinary skylight exposures, preventing condensation and minimizing the heat losses by diverting the heating medium (air) away from these exposures are the outstanding accomplishments of this installation.

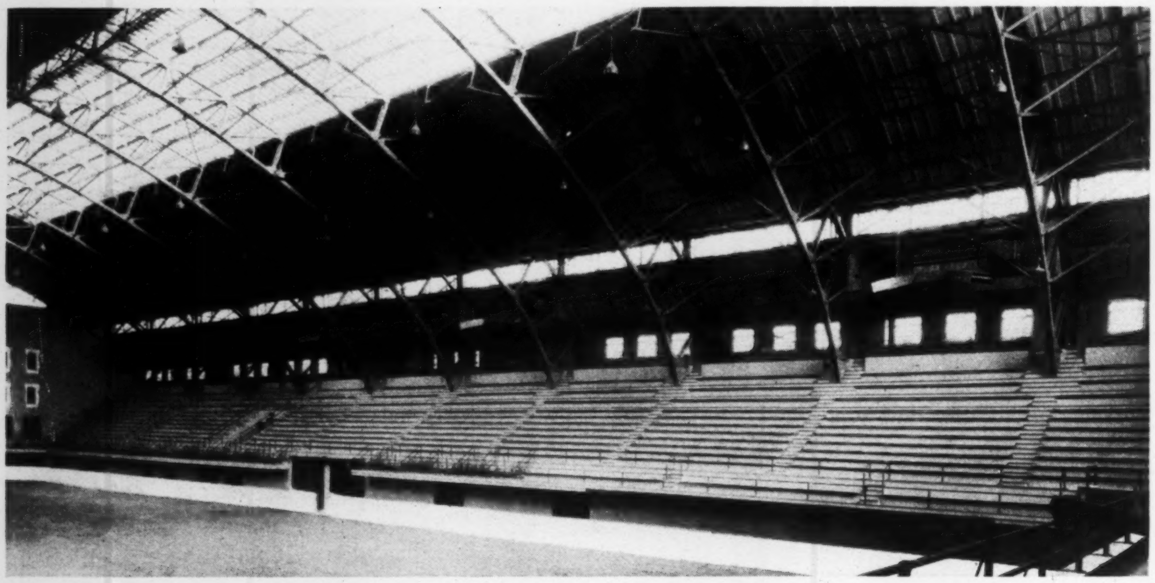

FIG. 2—THE UNITS ARE FIXED IN THE TRUSS SPACE IN THE ALTERNATE BAYS

"Open for Discussion"

Recent Article on Bourdon Tube Gages Discussed

By J. O. Tragard*

TWO of our staff, including the writer, have read the most interesting article on gages.[1] We give below our critical comments, which are offered in the helpful spirit.

(2) Under "Commercial Types of Gages" is listed the various sizes made. There are several sizes in this list which this company does not make; also, we manufacture 4-in., 5½-in., and 8-in., which are not listed. We believe these latter sizes are common to other manufacturers also. Standard dial finish with us is dull silver rather than dull white. Hydraulic gages are furnished with ½-in. hydraulic pipe threads.

(3) Under "Selecting the Proper Size of Gage" it is stated that the larger the dial size the easier it is to read and greater accuracy can be obtained. Instead of greater accuracy, it would be preferable to say that closer readings can be obtained. To state greater accuracy can be obtained should apply only to high pressure instruments which can be furnished more accurately in larger cases because larger springs may be used.

(4) Under "Selecting the Style of Gage," it is stated that where the cost must be kept to the minimum, iron case gages will *often* be found the lowest in price; we believe that iron case gages are universally cheaper.

(5) Under "Vacuum and Compound Gages"—the last paragraph—it is implied that steel springs are not used except for pressures of 1,000 lb. and over. As a matter of fact, many gage manufacturers have standardized on steel springs at 600 lb. and up and at least one has standardized on them down to all ranges above 400 lb., furnishing them on order as low as 0-60 lb.

(6) Under "Double Spring Gages" mention might be made of single tube gages which have a shortened tube to allow self-draining and thereby prevent freezing. One model also has a hardened steel movement for applications where extreme vibration or pulsation is present. Use of a hardened steel movement keeps the weight of the movement at a minimum, reducing friction and inaccuracy.

(7) Under "How Heat of Steam Affects Gages," it is stated that a gage tube should not be exposed to a temperature exceeding 150 F. Steel spring gages for temperatures as high as 300 F are sold, with the joints either welded or silver soldered. They will resist melting to a higher temperature.

(8) Under "Correcting Gage for Water Column," the article more or less encourages gage users to remove the indicating pointer from a gage to correct for water column. Usually, gage purchasers are not equipped to recalibrate the gage correctly after taking off the pointer. It often happens too, that the mechanic does not properly restake the pointer and consequent inaccuracy ensues. There are, of course, large users who understand the gage construction perfectly and who are

---

* Sales engineer, The Foxboro Company, Foxboro, Mass.

[1] "How to Install and Care for Gages of the Bourdon Tube Type," by John A. Masek. HEATING, PIPING AND AIR CONDITIONING, September, 1931. Page 727.

equipped to perform this operation. In further reference to this same paragraph, a screw on the pointer is referred to; this is known as a micrometer screw. The adjustment on this is, however, highly limited—generally to about 1 per cent of the total scale in either direction.

(9) Under "Danger of Over Pressure," the last paragraph, the use of a maximum hand on pressure gages is referred to. It has been the writer's experience that this maximum hand either creates a drag on the pointer, if fastened too tightly, or else bounces above the maximum pressure reached, if fastened too lightly.

Author of Article Replies

The kind criticism of my article is appreciated; it is a pleasure to note the interest shown. In writing the article, no attempt was made to refer to one manufacturer, but to make it general as possible. As a further caution, Navy Department specifications were referred to with the thought of obtaining representative standards. They, too, found difficulty in making their specifications standard for all makes as indicated by the following from their foreword. "No pretension is made that the specifications are perfect, but they are offered as providing for gages conforming to present engineering and manufacturing practice."

Referring to paragraph *2* above, gage dial sizes can be obtained in dimensions ranging from 1½ to 16 in. in diameter. Those mentioned in my article are those thought to be more or less standardized by usage. Gages with a range in pressure of 1,000 lb. are generally classed as hydraulic and furnished with ½-in. pipe thread connections.

A correct phrasing of words and interpretation is made for "Selecting the Proper Size of Gage," in paragraph *3*. It is also true that iron case gages are universally cheaper, as noted in paragraph *4*.

Steel bourdon tube springs (paragraph *5*) can be fabricated for any pressures but mention was made and intended for general standards. Manufacturers have used various metals in the construction of gage bourdon tubing since Bourdon of France invented and manufactured pressure and vacuum gages before 1850. The fundamental principle is still the same and is used by every gage manufacturer.

One manufacturer furnishes a gage with a special steel bourdon tube and special steel movement, so as to give the gage life and accuracy. Other manufacturers are using a similar construction with a movement made of an alloy. These movements are usually intended for service that is particularly heavy.

Whether a gage is made of bronze, steel or a special alloy, the writer believes that if used on steam at any pressure a siphon to keep live steam from the bourdon tube should be installed. Metal will lose its resiliency—if only slightly—with the application of temperature and time. On oil refining processes operating at pressures of 200 lb. and temperatures of 900 F, cooling legs are often provided to protect the gage.

Referring to paragraph *8*, if a method of removing a pointer with two screw drivers as shown in Fig. 1 were advocated, this would be wrong. However, the writer feels that the maintenance departments of many plants are capable of taking off the indicating pointer with a hand jack or puller, and replacing it. Information of this kind is valuable to the engineer in an isolated plant.

It is true that some types of maximum pointer indicating hands create a considerable drag on the indicating pointer. Other types are so aranged that they will move with practically little friction and will remain in place. Some hydraulic press manufacturers have special maximum indicating pointers hidden inside of the gages furnished on their hydraulic presses.

An interesting case is recalled where this device saved the day for one concern. A certain purchaser of a press operated it on over-pressure most of the time. This was kept up until the press went to pieces, causing damage amounting to several thousand dollars. The owner sued the manufacturer, who suspected over-pressure, had the gage brought in court, and opened up. The inside pointer indicated a pressure far beyond the pressure designated for that press.

The subject of gages is an old one, but there is little published material. Gages have often been "taken for granted" but careful study of their operation and applications is a mighty important matter to the engineer.—*John A. Masek.*

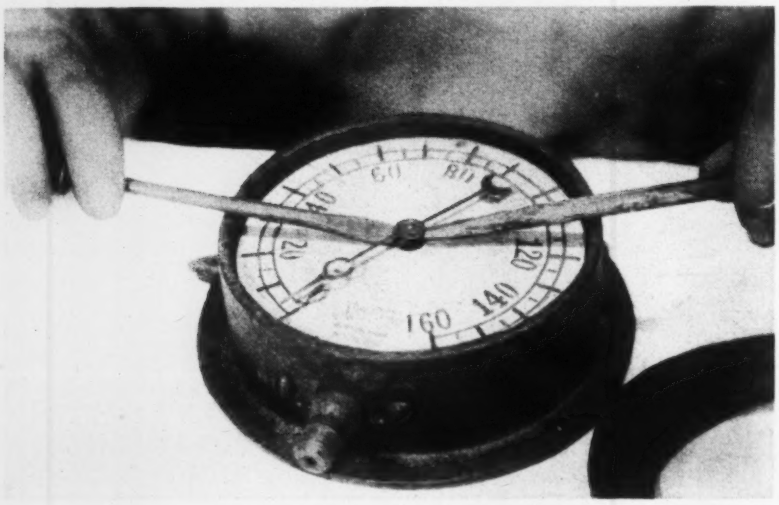

FIG. 1—A GAGE POINTER SHOULD NOT BE REMOVED IN THIS MANNER. HAND JACKS MADE ESPECIALLY FOR THE PURPOSE ARE AVAILABLE AND SHOULD BE USED WHEN NECESSARY BY THE EXPERIENCED MAINTENANCE MAN

Heating·Piping
and Air Conditioning

1895

American Society *of* Heating *and* Ventilating Engineers
JOURNAL SECTION

NOVEMBER *1931*

Material in this Section prepared by Publication Committee: W. A. Rowe. *Chairman;* E. O. Eastwood, and F. D. Mensing A. V. Hutchinson, Manager of Publications

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Some Fundamental Considerations of Corrosion in Steam and Condensate Lines

By R. E. Hall[1], Pittsburgh, Pa., and A. R. Mumford,[2] New York, N. Y.
NON-MEMBERS

THIS paper presents the results of an extensive investigation, covering a period of one year, into the factors influencing corrosion in steam systems. The investigation includes a study of the raw water and the production, the distribution, and the utilization of steam, with reference to heating systems and appliances only. Data are presented in full showing the extent and constitution of deposits in the heating systems of a large office building and a large hotel in New York City. These deposits were found to originate from the action of corrosion and were found to contain little or no evidence of carryover from the boilers. The cause of the excessive corrosion was found to be the inleakage of air into the vacuum return system. The amounts of oxygen and carbon dioxide associated with the steam are shown to be of insignificant importance as corrosive agents by the application of the Law of Henry and Dalton. The cause of corrosion trouble in heating systems can, therefore, be sought in the operation and, to a small extent, in the design of the systems and not in the quality of steam used if that quality equals that encountered in these studies.

Solvent Action of Water in the Steam Heating Cycle

Water is universally a solvent of the substances with which it contacts. Some substances, such as sugar, dissolve readily in water. Others, such as limestone, dissolve far less readily but nevertheless are dissolved by water. Since one of the purposes of this paper is to specify available means of defense against deleterious effects of the solvent action of water, attention will be directed first to certain laws of solubility and to certain mechanisms of reactions, a knowledge of which is requisite for the sagacious construction of adequate defense against these effects.

A compound substance may dissolve without chemical reaction with the water or with chemical reaction. Neglecting such chemical actions as are characterized by the acquisition or loss of so-called water of crystallization, most compounds dissolve in water without chemical reaction. The process of dissolution in such cases is characterized by a physical division of the crystals into molecules which disperse themselves through the water. Such compounds are further subdivided in solution by the breaking up of molecules into parts called ions. This process is called dissociation and each compound dissociates to a definite degree which is specific for that compound and which varies from a value of zero to one of almost one hundred per cent for different compounds in saturated solutions. The degree of dissociation for a particular compound is called the dissociation constant and this numerical value is the product of the equivalent concentrations of the two or more ions formed. If a substance enters into chemical reaction with water (other than actions involving water of crystallization) the reaction is between ions and the solution of the substance involves chemical changes which cannot be reversed under practical conditions whereas the physical changes in dissolution of the first class can be reversed by evaporating the water.

Solution of Ordinary Crystalline Compounds

When a small amount of sugar or salt is added to water the sugar or salt disappears more or less rapidly depending upon the amount of stirring. If the addition of sugar or salt is continued the water eventually becomes saturated and in this condition no further visible solution of the material will take place. However, although the total quantity of sugar or salt molecules in solution has become constant, the personnel is constantly changing because some molecules are continuously dissolving and an equal number of others are just as rapidly returning to the visible crystals.

The quantity of any substance which will dissolve in a given quantity of water is a measure of the specific solubility of the substance. The specific solubility of each substance is not only different from that of all other substances but the solubility of each changes in a different degree with the temperature of the water. Some substances become more soluble as the temperature rises while others become less soluble under like conditions. The action of each individual substance, when placed in contact with water, must be studied to learn the properties of the substance and no predictions of solubility can be made without such studies.

Dissociation of Water

Pure water contains not only molecules of water (H_2O) but separated parts of these molecules called ions (H^+, and OH^-). In pure water the amounts of hydrogen ion (H^+) and hydroxyl ion (OH^-) are equally small and, as was the case in a saturated solution, new molecules are continually splitting up at a rate equal to that at which hydrogen ions are combining with

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[1] Director, Hall Laboratories, Pittsburgh, Pa.

[2] Research and Design Engineer, New York Steam Corporation.

For presentation at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932.

hydroxyl ions. This condition of equilibrium is represented by the equation.

$$\text{Water} \rightleftharpoons \text{Hydrogen ion} + \text{Hydroxyl ion.}$$

Under conditions of equilibrium the product of the equivalent concentrations of hydrogen-ion and hydroxyl-ion is a constant. This constant is represented by the following equation for conditions at room temperature.

$$(H^+) \times (OH^-) = 1 \times 10^{-14}$$

The importance of this dissociation constant cannot be over-estimated in any study of corrosion. Any increase in the concentration of either ion must cause a decrease in the concentration of the other ion if the product is to remain constant. Consequently, if, for any reason, it is desirable to reduce the concentration of hydrogen ion such reduction can be readily accomplished by increasing the concentration of hydroxyl ion.

For convenience a number representing the logarithm of the reciprocal of the hydrogen-ion concentration is used to define the relation of (H^+) and (OH^-) in any solution. This number is called the pH value. A pH value greater than 7 means that the solution contains more (OH^-) than (H^+) and is therefore alkaline, whereas a pH value less than 7 means that the solution contains more (H^+) than (OH^-) and is therefore acid.

Solution of Metals

Because iron is the material of which most pipe is made, the mechanism of its solution in water will be discussed first. When iron is placed in water, it replaces some of the hydrogen ion, and goes into solution. In this process the hydrogen-ion concentration is reduced and more water ionizes to make up the deficiency. Because for each additional molecule of water which becomes ionized an additional hydroxyl is formed, the hydroxyl-ion concentration is continuously increasing during this process and because the product of the concentrations of (H^+) and (OH^-) must be a constant, a point is reached at which the concentration of hydrogen ion is so small that iron is no longer able to replace it and the solvent action of water on iron ceases. The chemical change involved in the solution of iron has, for one of its products ferrous hydroxide $(Fe\ (OH)_2)$ a soluble salt of iron. This compound ionizes to such an extent that the pH value of a solution of ferrous hydroxide is 9.6 and at this pH value an effect similar to *back pressure* prevents the formation of more ionic iron and thus prevents the solution of metallic iron. Whether ferrous salt is in solution or not, if the pH value is 9.6, because of the presence of sufficient hydroxyl ion, metallic iron cannot dissolve.

If, instead of permitting the dissolution of iron to build up the hydroxyl-ion concentration by using up the hydrogen ion, we increase the concentration of hydroxyl ion by adding caustic soda until the pH value of the solution is 9.6, we attain the same result, namely, prevention of the solution of iron.

If, on the other hand, we increase the concentration of hydrogen ion by adding sulphuric acid, and thus replenish the supply perhaps many times faster than the solution of iron would normally use it up, we increase both the rate of reaction and the quantity of iron which dissolves. Solution of the iron will continue until the concentration of the hydrogen is so small that the iron is no longer able to effect replacement.

The solvent action of water on a metal is a function of the hydrogen ion concentration of the water, and the power of the metal to replace hydrogen ion. The power of each metal to replace hydrogen ion in water differs from that of all other metals and the metals are usually listed in the order of the power. Of the common metals, copper, nickel, tin and lead have less power and aluminum and zinc have a greater power than iron to replace the hydrogen ion in water. The rapidity with which each metal replaces the hydrogen ion in water can be varied by placing in contact with the metal another metal with a more or less widely different power. The net difference in power, of the two contacting metals, expressed in volts is the force leading to the dissolution of the metal with the greater power and the arrangement is an electrolytic cell.

If instead of a different metal, an alloy of the principal metal is used the same action, although to a lesser degree, takes place. Such alloys or impurities are introduced in manufacturing but, during the past years, manufacturers have successfully reduced the danger of failure ascribable to the local segregations or inclusions of foreign material in their product by adjusting the process to assure uniformity of composition.

Mechanical strain introduced by working the metal is sufficient to set up an electrolytic cell at such points as nipple threads and bends and thus accelerate the normal rate of corrosion. The basic reaction is the same in all cases and the inhomogenity or strain is superimposed as an accelerant.

The apparent contradiction between the excess power of aluminum and zinc to replace hydrogen ion and the use of these metals in exposed conditions in preference to iron is answered by the presence of protective films. Both zinc and aluminum are protected by a thin film of oxide which segregates the metal from any contact with water and this film, if broken by a scratch or otherwise, is usually rebuilt by the corrosion process itself. If such a protective film, which constitutes the main defense against all corrosion of all types, is not rebuilt, corrosion occurs.

Iron has not the property of building a protective impermeable film on its exposed surface, except possibly in a solution having the correct pH value. A film of iron oxide is permeable both to gas and water and consequently offers no protection to the metal underneath. Iron in any form is the same and its power to replace hydrogen ion from water is nearly enough equal in all forms so that any differences in rates of dissolution ascribable to its form are of minor and not major importance. Unless aided by the establishment and maintenance of the requisite pH value in the contacting water no one of the forms is more or less able to protect itself by the formation of an impermeable protective coating. If, under like circumstances, real differences in the rates of corrosion of the different forms occur, it is due to other factors than the metal. Thickness of metal, in one form, results in a longer life under some conditions and adventitious protective coverings or inclusions may increase the service life. The effect of such coverings is excellent if they are everywhere interposed between the metal and the water but they are not rebuilt by the corrosive action and once the water passes them corrosion will proceed until the next covering layer is reached. Because such protection is adventitious and not of orderly planning its effectiveness can only be estimated by the average of many observations and does not lend itself to prediction for any specific case. It can be concluded, therefore, that the service conditions are the dominant factors in determining the service life of iron or steel and that differences of internal conditions are merely subsidiary to the main factors.

Of the common metals, lead, zinc, tin, nickel, aluminum and chromium have the property of interposing an impermeable layer of the metallic oxide between the metal and water. This layer is self-built and constitutes the principal reason for the resistance to corrosion. If such metals are successfully alloyed with iron the film forming property is retained in the alloy and, if the content of the film rebuilding metals is sufficient, the iron itself is protected from the solvent action of water. The film forming property is of major importance and overshadows such differences as may exist in power to replace the hydrogen ion in water. To alloy with iron a metal of less power to replace hydrogen ion, which metal does not have the film forming property, is usually not sufficient to make service conditions a minor factor in service life. Alloy steels which have properties of a nature to render them dominant over the service conditions to which they are subjected are not usually economical enough for pipe service.

The magnitude of the solvent action of pure water alone on metal is small, but other factors which we will consider, accelerate the solvent action, and multiply and magnify the deleterious results so that they become major considerations in the economical operation of the system.

It is necessary in any study of the resistance of various metals to corrosive action to emphasize service conditions as the major factor in operating life. The adventitious character of many of the differences in resistance to corrosion makes it very risky to predict performance unless the basis of the prediction is the history of service life in a large number of installations under like conditions.

Gases of the Atmosphere: Law of Henry and Dalton

Oxygen

Of the many gases in the atmosphere only oxygen and carbon dioxide will be considered because the others are of negligible importance to the solvent action of water on iron.

Oxygen gas dissolves in water when molecules of the gas penetrate the surface of the liquid and finally become distributed throughout its depth. Temporarily those layers of water near the surface will contain more oxygen molecules per unit volume than lower layers but the motion of the molecules will gradually disperse them through the liquid volume and, indeed, some of them will escape from the liquid at its surface and reenter the gaseous phase. When equilibrium is established the number of molecules of oxygen leaving the surface of the water is the same as the number entering the water from the gaseous phase and the water is saturated.

Obviously the number of molecules of oxygen entering the water depends on the number of such molecules in the gaseous phase and therefore the number of molecules of oxygen in solution will depend on the same factor. A reduction in the number of molecules of oxygen in the gaseous phase may be obtained by reducing the total gas pressure or by diluting the oxygen with some other gas such as nitrogen keeping the total gas pressure constant.

Partial Pressure

If a gas is diluted with another gas and the total pressure is kept the same, that part of the pressure due to the presence of the original gas is reduced by the partial pressure of the diluent gas. Thus in the atmosphere the total pressure is about 15 lb per square inch and this pressure is made up principally of the partial pressures of oxygen, nitrogen and carbon dioxide. The partial pressure of each constituent is in proportion to the ratio of the volume of each constituent to the total volume. In the case of the atmosphere, which is 21 per cent oxygen, 0.03 per cent carbon dioxide and the rest several inert gases, the partial pressure of oxygen is 3.15 lb per square inch and of carbon dioxide 0.045 lb per square inch.

Law of Henry and Dalton

From these considerations, it follows that the concentration of a gas such as oxygen dissolved in a liquid such as water is proportional to its partial pressure (or concentration) in the contacting gaseous phase. This generalization is known as the Law of Henry and Dalton, and it will be so designated in future references.

The importance of the Law of Henry and Dalton to this study of steam for heating purposes is in the fact that it enables computation of the amount of any gas which can be dissolved in water or condensate when the partial pressure of the gas in the contacting gaseous phase is known. The fact that only dissolved gases can have any influence on corrosion emphasizes the usefulness of the Law of Henry and Dalton. In the design of apparatus for the removal of dissolved gases from water, the Law of Henry and Dalton has been used as the guiding principle and such apparatus endeavors to reduce the partial pressure of the dissolved gases in the gaseous phase to as low a value as is practicable.

The specific solubilities of oxygen and carbon dioxide differ. Water at 60 F in contact with pure oxygen at 15 lb per square inch pressure is saturated when it has dissolved 34 milli-liters of oxygen per liter of water; in contact with pure carbon dioxide at the same pressure, water dissolves 1000 milli-liters of CO_2 per liter of water. In contact with air, under like conditions, $0.21 \times 34 = 7.1$ ml/liter of oxygen dissolves at saturation, and $0.0003 \times 1000 = 0.3$ ml/liter of carbon dioxide.

The rate of solution to a point of saturation depends on the surface of water exposed to the gas and the distance the dissolved molecules of the gas must travel to produce homogenity of distribution. The greater the surface the greater the area exposed to impingement by the gas molecules and the shallower the vessel the shorter the distance each dissolved molecule must travel. Conversely the rate of degasification of water depends on exactly the same factors as well as the partial pressure of the gas in the contacting gaseous phase.

The specific solubility of a gas changes with temperature but the law of proportionality of solubility in the liquid to partial pressure of the gas holds true regardless of changing specific solubility values. At 212 F and a partial pressure of oxygen of 15 lb per square inch, the solubility of oxygen is 17 ml/liter; but if the partial pressure of oxygen is 1.5 lb the solubility at this temperature is only 1.7 ml/liter. Regardless of specific solubility values, the solubility of a gas is zero if its partial pressure in the gaseous phase in contact with the liquid is zero.

TABLE 1—SOLUBILITY OF THE OXYGEN OF THE AIR IN WATER

When Partial Pressure of Water Vapor + Partial Pressure of Air = 14.696 lb

TEMPERATURE (DEG FAHR)	VAPOR PRESSURE OF WATER (LB PER SQ IN.)	PARTIAL PRESSURE OF AIR (LB PER SQ IN.)	SOLUBILITY	
			PURE OXYGEN ML./L (32 F, 14.696 LB PER SQ IN.)	OXYGEN OF AIR 21% O_2 ML./L (32 F, 14.696 LB PER SQ IN.)
32	0.0887	14.607	48.89	10.20
40	0.1217	14.574	43.50	9.05
50	0.1780	14.518	38.02	7.9
60	0.2561	14.440	33.8	7.0
70	0.3628	14.333	30.5	6.2
80	0.5067	14.189	27.5	5.6
90	0.6980	13.998	25.4	5.1
100	0.9487	13.747	23.1	4.65
110	1.274	13.422	22.2	4.27
120	1.692	13.004	21.0	3.9
130	2.221	12.475	20.0	3.6
140	2.887	11.809	19.46	3.25
150	3.716	10.980	18.50	2.9
160	4.739	9.957	18.00	2.6
170	5.990	8.706	17.60	2.2
180	7.510	7.186	17.50	1.8
190	9.336	5.360	17.40	1.3
200	11.525	3.171	17.15	0.78
210	14.123	0.573	17.05	0.15
212	14.696	0.000	17.00	0.00

Carbon Dioxide

Carbon dioxide in steam or condensate in general follows the same rules that govern solubility of oxygen. The carbon dioxide content in any water may be present as dissolved carbon dioxide (CO_2), as bicarbonate radical $(HCO_3)^{-}$ and as carbonate radical $(CO_3)^{--}$ associated with a positive metallic radical. Since the bicarbonate radical readily decomposes to form dissolved and finally gaseous carbon dioxide (CO_2), it is frequently referred to as carbon dioxide in the half-bound condition.

Bicarbonate Formation

It is pertinent at this point to discuss the mechanism of accumulation of available carbon dioxide in the raw water.

When carbon dioxide dissolves in water, some of it reacts with water thus:

$$\underset{CO_2}{\textit{Carbon Dioxide}} + \underset{H_2O}{\textit{Water}} \rightleftharpoons \underset{H_2CO_3}{\textit{Carbonic Acid}}$$

The carbon dioxide thus formed ionizes to some extent as follows:

$$\underset{H_2CO_3}{\textit{Carbonic Acid}} \rightleftharpoons \underset{H^+}{\textit{Hydrogen Ion}} + \underset{HCO_3^-}{\textit{Bicarbonate Ion}}$$

By so ionizing, it raises the hydrogen-ion concentration of the water. As the raw water passes along its course, it comes in contact with limestone ($CaCO_3$) and dissolves some of it. The limestone is relatively insoluble, but the small amount dissolved ionizes thus:

$$\underset{CaCO_3}{\textit{Limestone}} \rightleftharpoons \underset{Ca^{++}}{\textit{Calcium Ion}} \textit{ and } \underset{CO_3^{--}}{\textit{Carbonate Ion}}$$

With carbonic acid present, the next thing to occur is as follows:

$$\underset{CO_3^{--}}{\textit{Carbonate Ion}} \textit{ and } \underset{H^+}{\textit{Hydrogen Ion}} \rightarrow \underset{HCO_3^-}{\textit{Bicarbonate Ion}}$$

Thus, since they unite to form an additional bicarbonate ion, both hydrogen ion and carbonate ion are removed from the water. When this has occurred, the water is no longer saturated either with limestone or carbonic acid, more of them dissolve, and so gradually the water accumulates a quantity of bicarbonate. Because all reversible chemical reactions will continue until a state of equilibrium is reached when the tendency for the reaction to proceed in either direction is equalized, the introduction of a secondary reaction which removes one or more of the products of the primary reaction upsets the balance of the original equilibrium. In the case of limestone, the secondary reaction and the primary reaction must both reach equilibrium before the dissolution of limestone ceases. In such cases as the solution of iron, where one product of a secondary reaction is so insoluble that it can no longer have any practical effect, the original reaction continues until one or more of the ingredients is used up.

Thus solvent action along its course brings the water to the point of steam generation with a content of dissolved material the amount and character of which reflect the geographic locus of the drainage area whence it is derived. Table 2 illustrates this fact. Column 3 shows that in practically all North American surface waters the accumulation of bicarbonate goes on. From Columns 1 and 2 it is seen that the bicarbonate, or available carbon dioxide, in the Croton water is more than twice that in the Catskill; but by comparison with Col. 4 it appears that Croton water has considerably less than half the bicarbonate in Detroit city water.

Results of Simultaneous Solvent Action of Water with Atmospheric Gases on Metals

Carbon Dioxide

At *pH* values of 9.6 or greater, iron has practically no tendency to dissolve in water. At *pH* values less than this iron does dissolve and the tendency to dissolve increases as the *pH* value decreases.

Hence the dissolution of carbon dioxide in water accelerates the dissolution of iron therein, because the carbon dioxide on dissolving decreases the *pH* value of the water. In the same way the addition of any acid, as sulphuric acid, to the water, decreases the *pH* value therein and accelerates the dissolution of iron. The dissociation constants of acids vary with the acid and, in general, are an indication of the strength of the acid. Carbonic acid, formed by the solution of carbon dioxide in water, is a weak acid and dissociates but little in comparison with the strong acid, sulphuric. Carbonic acid, however, dissociates to produce a greater concentration of hydrogen ion than water and for this reason the *pH* value is lowered by the solution of carbon dioxide and the dissolution of iron is accelerated.

Although dissolved carbon dioxide has the same effect in accelerating the dissolution of iron, as has an acid like

TABLE 2—DISSOLVED SOLIDS IN THE CROTON AND CATSKILL WATER AND OTHER CHARACTERISTIC WATERS
PARTS PER MILLION

	1	2	3	4
	CROTON	CATSKILL	RELATIVE AVERAGE COMPOSITION NORTH AMERICAN SURFACE WATERS	CITY OF DETROIT
Bicarbonate (HCO_3)	36.6	15.9	67.89	96.5
Sulphate (SO_4)	8.2	3.2	15.31	16.4
Chloride (Cl)	5.0	1.3	7.44	7.0
Nitrate (NO_3)	0.6	0.6	1.15	.....
Silica (SiO_2)	9.0	1.0	8.60	2.8
Iron (Fe)	.....	.....	0.45	0.6
Calcium (Ca)	9.6	5.2	19.36	26.2
Magnesium (Mg)	3.5	0.8	4.87	7.4
Sodium (Na)	3.5	1.2	8.50	4.4
Total Solids	57.4	21.1	99.0	112.3
Available Carbon Dioxide (CO_2)	26.4	11.5	49.0	69.5
Dissolved Unbound Carbon Dioxide at saturation [a] at 60 F	0.6	0.6	0.6	0.6
Dissolved Oxygen at saturation [a] at 60 F	10.0	10.0	10.0	10.0

[a] Saturated in contact with the atmosphere.

sulphuric, fortunately the quantity factor in the dissolution by dissolved carbon dioxide is much lower than that of sulphuric acid. Thus, when water containing sulphuric acid is in contact with iron, dissolution continues until the acid is entirely exhausted and the *pH* value has risen to 9.6. When dissolution occurs by reason of dissolved carbon dioxide, the *pH* value begins to rise immediately ferrous salt is present in solution, and the dissolving action is vastly retarded or stopped long before the quantity of iron dissolved is equivalent to the carbonic acid present.

Thus, though the dissolved carbon dioxide sets up in the water the low *pH* value that is accelerative to dissolution of the metal, there is no sustained dissolution thereof because the dissolving process itself supplies to the water the ferrous iron that increases the *pH* value and thus slows down the dissolving tendencies. It is axiomatic that at high concentration of carbon dioxide, more dissolution of iron must occur to effect this result, than at low concentration; and therefore corrollary that the concentration of carbon dioxide in the steam should be suitably restricted, so that any water condensed therefrom, and dissolving carbon dioxide therefrom in accordance with the Law of Henry and Dalton, should contain insufficient dissolved carbon dioxide to cause any substantial dissolution of the metal before a protective *pH* value is established in the water.

Because of the low quantity factor of corrosion by dissolved carbon dioxide limitation of the quantity in the steam to a value of 15 to 20 ppm. or thereabouts, is advisable, but advantages to be gained by any further dimunition in carbon dioxide content are at a cost incommensurate with their value. The figure 15-20 ppm. is mentioned as an arbitrary standard, because it is obtained without great difficulty and because experience and theory agree that this amount of carbon dioxide in the absence of oxygen cannot be held responsible for corrosive effects worthy of consideration.

Oxygen

As iron dissolves in water, ferrous hydroxide and hydrogen are formed. When oxygen is simultaneously present, it unites with ferrous hydroxide to form ferric hydroxide, and with hydrogen to form water. The chemical reactions are as follows:

$$2\,Fe(OH)_2 + 1/2\;O_2 + H_2O \rightarrow 2\,Fe(OH)_3$$
$$H_2 + 1/2\;O_2 \rightarrow H_2O$$

Ferric hydroxide is red like ordinary iron rust and is very insoluble. Thus dissolved oxygen, by uniting with the ferrous hydroxide, and changing it to insoluble ferric hydroxide, removes all dissolved iron from the solution and hence the water is as free to attack the metal as though it were freshly distilled pure water. Also, dissolved oxygen by uniting with the hydrogen removes it, and thus any presence of hydrogen does not interfere with the rapid further dissolution of iron. The fact must be emphasized that dissolved oxygen accelerates dissolution of iron by constantly removing the iron from solution, and for the same cause it sustains the dissolving of iron. Oxygen is used up in the process, and naturally, if there is only a limited supply of oxygen in the water it will finally be used up, whereupon ferrous hydroxide can accumulate, retard, and finally stop any dissolving action. But on the other hand, if the water is in contact with a gaseous source of oxygen, then as dissolution of the metal uses up the oxygen in the water, the gaseous supply of oxygen hastens to replenish the depleted supply in the water in accordance with the Law of Henry and Dalton.

Under such conditions there is no limitation to the amount of iron that can be corroded by any given amount of water. The process is continuous, water acting merely as a contact agent to bring together the iron and oxygen.

Carbon Dioxide and Oxygen

Carbon dioxide accelerates dissolution of iron by decreasing the *pH* value, and thus hastens replenishment of the ferrous iron removed from the water by union with oxygen. When the iron is thus steadily removed, the low *pH* value due to the carbon dioxide is steadily maintained, so that the rate of dissolution of iron remains high, and oxygen, by uniting with the iron and removing it from the water, is the sustaining factor of this high rate of dissolution. Hence, it follows that with both carbon dioxide and oxygen present, more rapid continuance of corrosion will occur than with either one singly; and again, if oxygen in gaseous form is in contact with the water to replenish, in accordance with the Law of Henry and Dalton, any oxygen used by uniting with iron, then the amount of corrosion that may occur at this high rate is unlimited.

Pure water, or water containing dissolved carbon dioxide to a very limited extent, exercises a dissolving action on the metal, but in so slight a degree as to be negligible in the great majority of cases. However, when oxygen is present as a sustaining factor in this dissolution, then the limitation of dissolution that will occur depends entirely upon the total oxygen available to combine with the dissolved iron and thus sustain the reaction.

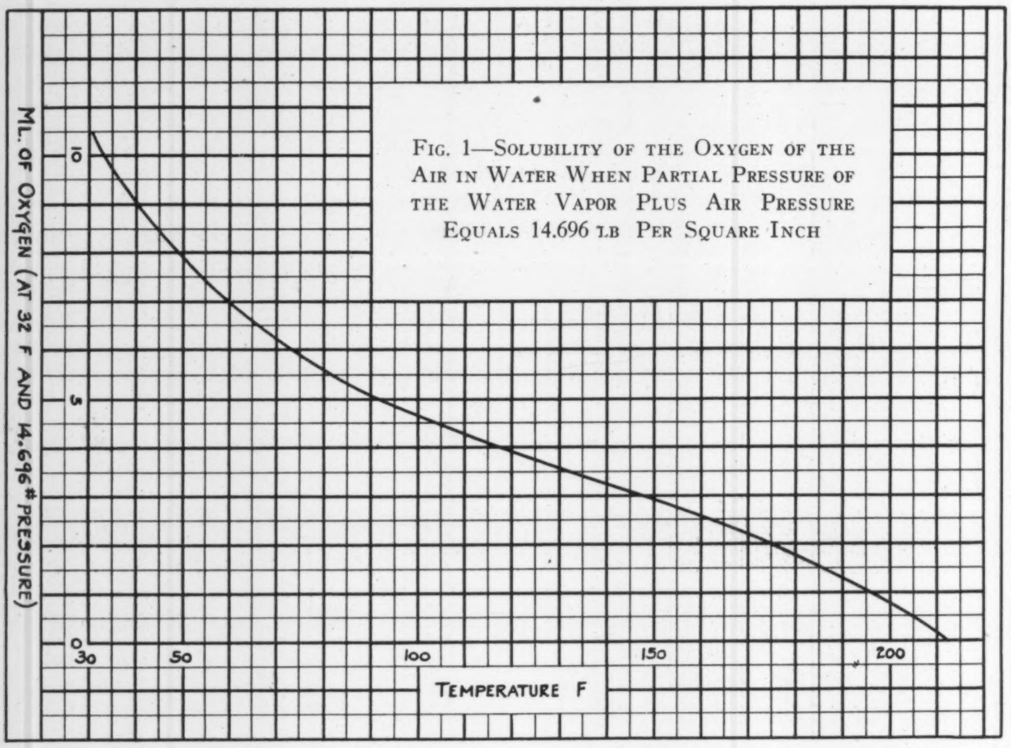

Fig. 1—Solubility of the Oxygen of the Air in Water When Partial Pressure of the Water Vapor Plus Air Pressure Equals 14.696 Lb Per Square Inch

Table 3—Temperature and Oxygen Content of Raw and Feed Water—Kips Bay Station

Date 1930	Raw Water		Feed Water	
	Temp. (Deg Fahr)	Oxygen ml./liter	Temp. (Deg Fahr)	Oxygen ml./liter
12/11	48	8.0	170	0.7
2/5	39	8.0	172	1.7
12/29	46	8.1	176	0.6
12/2	48	8.0	180	0.6
4/8	40	8.0	180	1.0
1/28	40	6.4	180	1.06
2/11	39	7.0	180	1.3
2/18	38	7.9	181	1.6
6/26	68	8.0	182	1.0
11/7	62	7.8	186	0.5
11/21	58	8.0	186	0.5
6/13	60	7.9	186	0.9
1/22	40	6.3	186	0.95
8/27	76	6.1	186	1.0
3/4	44	7.8	186	1.06
11/14	58	8.0	188	0.6
3/25	42	6.0	188	0.7
6/4	60	7.9	188	1.0
9/18	74	7.4	190	0.5
4/22	48	7.9	190	1.2
4/1	42	8.1	190	1.3
3/11	40	7.7	192	1.0
1/7	44	7.6	195	0.36
8/13	72	7.8	195	0.5
9/9	74	7.5	196	0.4
7/3	65	7.9	198	1.0
9/4	72	7.6	200	0.5
2/25	40	7.8	200	1.2
9/10	74	7.8	202	0.6
4/29	50	8.1	206	1.0
6/19	68	7.9	208	0.9
5/6	52	8.0	210	0.7
5/13	54	7.8	210	0.7
4/15	46	7.1	210	1.3

Analysis of Conditions in the Steam Generating, Distributing and Utilizing Systems

Problems of the Steam-Generating System

The steam-generating system must (1) develop steam sufficiently free from association with oxygen and carbon dioxide to meet the requirements of the remainder of the system; (2) establish and maintain conditions in the water such that its contacts with the surfaces of the generating system will not result in interruption of its continuity of operation, or effect deterioration; and (3) prevent entrainment by the steam of boiler water or solids, which by deposition in valves or lines in the distribution or utilizing systems, may cause trouble.

Separation of Associated Gases—Oxygen

If air-saturated water is heated in a vessel with restricted outlet, the oxygen content will decrease because the specific solubility decreases with rise in temperature and because the vapor pressure of water increases with temperature. As the vapor pressure of water increases the partial pressure of water vapor increases, and, therefore, the partial pressure of oxygen in the gaseous phase must decrease thus automatically decreasing the solubility of oxygen. In Fig. 1 the solubility of oxygen at increasing temperature and decreasing partial pressure, because of the increasing partial pressure of water vapor, is shown for the full range of water temperatures under atmospheric pressure. At 212 F the vapor pressure of water is equal to the atmospheric pressure so that the partial pressure of oxygen must be zero and, therefore the solubility of oxygen is zero.

Table 3 gives values of actual tests on feed water in an open, vented, feed-water heater. These values are sometimes larger, sometimes smaller than the values shown in Fig. 1 for the corresponding temperature. The rate at which water and low pressure steam are supplied to such heaters varies with the demand on the heater and with the supply of exhaust steam respectively. Obviously the time consumed by the water in passing through the heater varies and this variation is apparently sufficient to shorten, during some periods, the time below the minimum required by the molecules of dissolved oxygen to travel to the nearest liquid surface and escape into the vapor phase.

If the heater is provided with sufficient excess steam to maintain continuously the water temperature at 210 to 212 F, and to vent from the proper point in the heater, a quantity of steam sufficient to insure a partial pressure of oxygen as nearly zero as may be, then the oxygen content of the effluent water will not exceed 0.3 ml./liter and perhaps less. While 0.3 ml./liter is not as low an oxygen content as is guaranteed with some types of deaerating heaters (0.025 ml./liter) it is the authors' belief (because of the Law of Henry and Dalton) that this degree of removal is ample for the purposes of steam heating. The records of the generating system showing maintained heater-temperature, and of uninterrupted ample venting, constitute unimpeachable evidence as to the oxygen content of the steam it is furnishing.

Solution of the problem of oxygen-removal whether at the heater in the generating system, or in the utilizing system where inleakage of air is occurring, rests upon the fundamental factors of providing the very minimum of partial pressure of oxygen in gases making contact with the water, and providing a very short path for the dissolved oxygen molecules to the contacting surfaces. Practice must recognize the second, or time factor, although theory holds the first factor sufficient in itself for the removal of oxygen. The partial pressure factor at the heater is controlled by adequate supply of steam and venting; the time factor is controlled by spraying the water, or by passing it over trays in thin layers, to provide a short path of escape for the dissolved oxygen molecules. In the utilizing division there is only crude control of the partial-pressure factor and no control of the time factor. A pump may intermittently control the partial pressure of oxygen in the utilizing system but the rate at which oxygen is absorbed by the condensate when the steam is condensed far exceeds the rate at which oxygen is liberated by the main stream of condensate even though the partial pressure of oxygen in the contacting vapor phase is reduced.

Separation of Associated Gases—Carbon Dioxide

In the district steam plant and in the industrial plant with a large per cent of make-up, it is usually uneconomical to evaporate and effectively degasify all raw-water make-up in order to eliminate completely carbon dioxide. The slightly more complete exclusion of carbon dioxide by this process than by judiciously chosen treatment offers no particular advantages commensurate with the cost. It should be noted that the percentage of raw discriminately treated New York City make-up water has too little effect on the carbon dioxide content of the steam made therefrom to place the steam at any disadvantage for heating purposes as compared with steam made from water containing an equivalent amount of heating returns.

In waters which contain as much or more bicarbonate than that shown by Detroit city water the available carbon dioxide content can usually be considerably reduced by pretreatment with lime and soda. In such cases, the authors believe that lime-soda pretreatment represents the most economical solution for decreasing the available carbon dioxide, as well as to best prepare the water for boiler feed water. There is, however, only a slight gain to be realized by pretreatment of Croton water with lime and soda because the available carbon dioxide is reduced by reactions at the heater without pretreatment almost as much as it would be by pretreatment. The available carbon dioxide in the Catskill water is too low naturally to be improved by treatment.

No decrease of available carbon dioxide occurs with pretreatment by zeolite alone. The zeolite mineral effectively removes the calcium and magnesium from the water but does not remove the bicarbonate. Excessive quantities of bicarbonate in the feed water result in, among other things, excessive carbon dioxide in the steam and therefore subsidiary lime treatment and acid treatment, or the latter alone, are required together with all the accompanying niceties of control that distinguish acid treatment wherever used, when a bicarbonate water is treated with zeolite.

Table 4—Dissolved Solids in the Croton and Catskill Water and Other Characteristic Waters After Lime-Soda or Zeolite Treatment

Parts per Million

	Croton		Catskill		Rel. Average Comp. N.A. Surface Waters		City of Detroit	
	Lime-Soda	Zeolite	Lime Soda	Zeolite	Lime-Soda	Zeolite	Lime Soda	Zeolite
Hydroxide (OH).....	1.7	0.0	1.7	0.0	1.7	0.0	1.7	0.0
Carbonate (CO_3).....	18.0	0.0	18.0	0.0	18.0	0.0	18.0	0.0
Bicarbonate (HCO_3)..	0.0	36.6	0.0	15.9	0.0	67.9	0.0	96.5
Sulphate (SO_4).......	8.2	8.2	3.2	3.2	15.3	15.3	16.4	16.4
Chloride (Cl)........	5.0	5.0	1.3	1.3	7.4	7.4	7.0	7.0
Nitrate (NO_3).......	0.6	0.6	0.6	0.6	1.2	1.2	....	....
Silica (SiO_2).........	9.0	9.0	1.0	1.0	8.6	8.6	2.8	2.8
Iron (Fe)............	....	....	....	....	0.5	0.5	0.6	0.6
Calcium (Ca)........	6.0	2.0	6.0	2.0	6.0	2.0	6.0	2.0
Magnesium (Mg).....	0.8	0.5	0.8	0.5	0.8	0.5	0.8	0.5
Sodium (Na)........	15.0	17.9	10.3	5.5	20.2	35.0	20.0	45.7
Total Solids.........	64.3	61.2	42.9	21.9	79.7	103.9	73.3	122.5
Available Carbon Dioxide (CO_2).....	13.2	26.4	13.2	11.5	13.2	49.0	13.2	69.5

Table 4 shows the results of treating the waters of Table 2 with lime-soda and with zeolite. The effect of the pretreatment on the bicarbonate and therefore the relative effect on the carbon dioxide content of the steam is clearly evident.

As the temperature of a bicarbonate-containing water is raised, the tendency of the bicarbonate is to revert to the carbonate, and at the temperature of normal boiling reversion proceeds with considerable speed. If sufficient time were given under the conditions prevailing at the heater, reversion would be complete, and the only remaining available carbon dioxide in the water would be in the form of normal carbonate from which loss of carbon dioxide occurs considerably less rapidly than from the bicarbonate.

At the heater, therefore, and particularly under the conditions of maintained temperature and ample venting of steam, much of the bicarbonate will revert to carbonate with very considerable reduction of the available carbon dioxide passing on to the boilers. Jackson and McDermet (*J. Ind. Eng. Chem.* Vol. 15, pp. 959-61, 1923) have determined that by deaeration carbon dioxide loosely combined as bicarbonate is removed to the extent of about thirty per cent. That this is the case with Croton water is shown by the figures of Table 5, in which the maximum carbon dioxide found in the steam was 21.9 ppm. and minimum 15.6, while the general figure for available carbon dioxide in the feed water was 26.4 ppm. (Table 3, Column 1). Most of this decrease in available carbon dioxide is due to its separation at the heater; however, some small amounts separate in the form of economizer deposits and boiler sludge as illustrated by Table 6. Then again, all available carbon dioxide in the boiler water does not pass into the steam, as there is always some carbonate in the boiler water as illustrated by Table 7, which lists the average concentrations for the month of December, 1930, in a phosphate-treated boiler water from a large boiler operating at high ratings on Croton River water.

Discussion of water conditioning for the specific purpose of protecting the boiler surfaces is beyond the scope of this report, and reference is made to papers directed thereon: Mumford, *Combustion,* Vol. 1, pp. 43-46, Sept. 1929; Markson, *Combustion,* Vol. 1, pp. 27-30, April 1930; Hall, *Iron and Steel Engr.* Vol. 6, pp. 380-389, (1929); *Mech. Engr.,* Vol. 48, 317-327 (1926); *Trans. A. S. M. E.,* Fuels and Steam Power Sections, Vol. 50, No. 33, pp. 65-75, (1928).

Prevention of Entrainment of Boiler Water in the Steam

If boiler water is entrained in steam a quantity of dissolved or suspended matter is carried with the water which depends on the amount of water entrained and the concentration of solids in the boiler water. The distance from the steam generator to which such solids may be carried depends on the steam velocity and on the frequency of impingements which have a coalescing and separating effect and is of interest because of its effects in the distributing and utilizing divisions, particularly the latter, because deposits therein primarily due to other causes may be mistakenly ascribed to such carry-over.

In Table 8 is presented an analysis (No. 1) of sludge characteristic of that occurring in a phosphate-treated boiler water; and analyses (Nos. 2-7) of various deposits taken from the mains of the distributing system. That all are largely or in part derived from carry-over of boiler water in the steam is obvious because of the phosphate content for which no other source is possible. Some of them, notably Nos. 2, 4 and 7 in particular, show admixture of other material with the boiler sludge, as shown by the very considerable amount of silica they contain; and others, notably Nos. 4, 5, 6 and 7, by their magnetic quality, and their content of iron oxide, show accretion of corrosion products from some source—probably the trap, in view of the conclusions developed in the discussion of Division 2.

In Table 9 are presented analyses of several samples of condensate from various points in the system. Those from the street traps contained considerable suspended and dissolved solids; those taken from Building *A* contained but little dissolved and suspended solids, indicating that any carry-over of boiler water in the steam is largely removed in the traps on the distributing mains before the steam enters the building. In practically all cases, phosphate was present; and likewise iron. Further on, in the analyses of Table 12, will be found the relative quantities of iron oxide and phosphate in samples of deposits taken from various points in Building *A.*

TABLE 5—DISSOLVED GASES IN STEAM DERIVED FROM CROTON WATER

South Main

DATE	OXYGEN ML./LITER	CARBON DIOXIDE PARTS PER MILLION	*pH* VALUE OF CONDENSATE
April, 1930			
4	0.50	20.6	4.9
5	.45	21.9	4.7
8	.41	19.7	5.2
9	.38	18.6	4.9
10	.34	18.1	4.8
11	.46	19.8	5.0
14	.45	20.8	4.9
15	.34	18.1	5.2
16	.45	19.7	5.9
17	.41	19.7	6.0
		North Main	
August, 1930			
5	0.34	16.3	5.0
6	.32	15.6	5.0
7	.39	21.3	5.2
8	.35	16.3	5.2
12	.49	18.5	5.0
13	.42	17.3	5.0
15	.28	20.3	5.0
26	.35	17.5	5.0

TABLE 6—ELIMINATION OF CARBON DIOXIDE IN DEPOSITS AND SLUDGES—ANALYSES IN PER CENT

	SLUDGE FROM WATER DRUM	SCALE WATER DRUM	No. 3 ECONOMIZER DEPOSIT
Carbon Dioxide (CO_2)	4.75	14.41	14.83
Sulphur Trioxide (SO_3)	3.14	10.70	1.36
Phosphorous Pentoxide (P_2O_5)	24.04	5.26	17.73
Silica (SiO_2)	9.74	6.70	6.54
Ferric Oxide (Fe_2O_3)	3.10	2.84	1.24
Aluminum Oxide (Al_2O_3)	....	0.33	....
Calcium Oxide (CaO)	38.06	39.50	45.52
Magnesium Oxide (MgO)	10.38	15.75	7.70
Sodium Oxide (Na_2O)	....	2.80	....
Loss 105 C	0.45	0.65	0.30
Net Ignition Loss	6.15	0.14	4.80

TABLE 7—AVERAGE COMPOSITION OF BOILER WATER—PARTS PER MILLION

No. 5 BOILER—*December* 1930

Sodium Hydroxide ($NaOH$)	260
Sodium Carbonate (Na_2CO_3)	143
Sodium Phosphate (Na_3PO_4)	252
Sodium Sulphate (Na_2SO_4)	1450
Sodium Chloride ($NaCl$)	Undetermined

TABLE 8—BOILER SLUDGE AND DEPOSITS TAKEN FROM TRAPS ON DISTRIBUTION MAINS—ANALYSES IN PER CENT

	1	2	3	4	5	6	7
	BOILER SLUDGE No. 5 BOILER	GATE VALVE DEPOSIT 1ST AVE. MAIN DISTRIBUTING LINE	NORTH TRAP WEST SET. 622 1ST AVE.	SOUTH TRAP 37TH ST. E. OF 3RD AVE.	SOUTH TRAP 41ST ST. E. OF PARK AVE.	SERVICE TRAP ON 150 LB. HEADER AND *C* METER	SIX INCH VALVE 90 LB. HEADER
Sulphur Trioxide (SO_3)	1.3	Trace	0.6	0.3	0.6	0.8	1.9
Carbon Dioxide (CO_2)	1.6	included in Ignition Loss					
Phosphorous Pentoxide (P_2O_5)	31.5	29.9	33.3	7.6	27.1	19.2	9.8
Silica (SiO_2)	4.6	12.1	7.6	32.9	8.6	11.5	16.9
Ferric Oxide (Fe_2O_3)	0.6	1.2	8.1	39.3	27.3	41.0	17.6
Aluminum Oxide (Al_2O_3)			1.0	4.4	1.8	2.3	4.0
Calcium Oxide (CaO)	36.3	37.8	35.7	7.9	26.2	20.7	17.4
Magnesium Oxide (M_gO)	17.7	13.6	13.5	2.7	5.2	3.1	23.1
Loss at 105 C	2.7	0.5	1.4	0.8	1.2	1.0	2.8
Ignition Loss	4.3	5.7	4.1	7.2	2.0	1.6	12.8
	non-magnetic	non-magnetic	non-magnetic	magnetic	slightly magnetic	slightly magnetic	magnetic

TABLE 8A—DESCRIPTION OF SAMPLES OF TABLE 8

1. *Boiler Sludge*
   Tan powder, easily pulverized. Non-magnetic.
2. *Gate Valve Deposit, 1st Ave. Main Distributing Line*
   White deposit readily pulverized to impalpable powder. Non-magnetic.
3. *North Trap, West Set. 622 1st Ave.*
   Very soft, light gray powder, non-magnetic. Amount of deposit 40 grams.
4. *South Trap, 37th St. East of 3rd Ave.*
   Dark gray, soft powder, magnetic. Amount 30 gms.
5. *South Trap, 41st St. East of Park Ave.*
   Light brown powder, slightly magnetic. Amount, 10 gms.
6. *Service Trap on 150-lb Header and C Meter.*
   Soft mud on sides and bottom of trap. When dry, light brown, slightly magnetic powder. Amount not recorded.
7. *Six-Inch Valve, 90-lb Header*
   Light tan, scaly deposit on downstream side of gate. Very small amount, magnetic light gray, soft. Amount, about 1.5 gms.

The high content of iron oxide and low content of phosphate in these deposits is indubitable proof that their major source is not boiler-water sludge but the dissolution of the piping system of the building itself, and its transformation to the red oxide by the inseepage of the atmospheric gases into that system.

While everything, therefore, points to the fact that in this case carry-over in the steam has little or no relation to the problem of deposits beyond the street meter, the prevention of carry-over has been and is vitally necessary to the generating system for its own economic operation, and totally independent of its relations to the utilizing system. The reasons need not be enumerated here, but are inherent in conditions in the generating station, and in the distributing mains.

As the scope of this report does not permit general discussion of foaming and priming, reference is made to publications thereon. (Hall, *N. E. L. A.* Pub. No. 278-81, *Journal American Water Works Association,* Vol. 21, pp. 79-100; *Trans. A. S. M. E.* Fuels and Steam

TABLE 9—ANALYSES OF CONDENSATE TAKEN FROM STREET MANHOLES AND TRAPS FROM BUILDING A—PARTS PER MILLION

LOCATION	DATE OF SAMPLING	SUSPENDED MATTER	DISSOLVED SOLIDS	PHOSPHATE PO_4	IRON
Street trap, 622 1st Ave	9/8/30	137[a]	456[b]	present	present
Street trap, 37th St. E. of 3rd Ave	9/8/30	419	30	present	present
Street trap, 41st St. E. of Park Ave	9/8/30	9717	56	present	present
Trap at Meter	Shortly prior to April, '30	88	36	25	not recorded
Meter in Building	4/1/30	Total Solids 3 ppm.	....	none found	present
Sample by cracking valve on trap at main 90-lb. header	4/1/30	Total Solids 1.6 ppm.	....	trace	none found
Body of *C* Meter in Bldg	8/5/30	1	7	none found	present
Trap of *C* Meter in Bldg.	8/8/30	1	5	none found	present
Trap of main 90-lb. header	8/7/30	4	12	present	present
8-in. H.P. riser from 90-lb. header	8/6/30	0	3	none found	present
Inlet side trap at heel of 8-in. riser	8/25/30	1	3	present	present
Return hot water heater on 29th floor. Before trap	8/12/30	1	9	present	present
Return hot water heater on 29th floor. After trap	8/13/30	35	14	present	present
Kitchen Uptown Club, 26th floor. Returns from Coffee Urns	8/15/30	1	4	present	present
19th floor. Intermediate System	4/1/30	Total Solids 2.6 ppm.	....	none found	none found
Low pressure returns before vacuum pump	8/25/30	0	18	none found	present

[a] Result is low because of colloidal nature of some of the suspended matter making it impossible to obtain perfectly clear filtrate.
[b] Result is high because of colloidal nature of some of the suspended matter carried into filtrate.

Power Sections, Vol. 50, No. 33, pp. 65-75.) Discussion of the influence on the Kips Bay boilers of boiler-water alkalinities, cleanliness, or concentrations in their relation to carry-over; or the mechanical features of boiler baffling, installation of separators and continuous blowdown, control of water levels, drainage of steam drum, is a matter of record in the publications of Mumford (*Combustion,* September 1929, and *Trans. A. S. M. E.* Fuels and Steam Power, Vol. 51, No. 22, pp. 363-374) and Markson (*Combustion,* April 1930).

The problem of carry-over is receiving attention to a degree increasing with the capacity of the steam generating units. The knowledge on which the solutions of the problem of carry-over could be based must be built up by the engineers responsible for the operation of these new generators because the changes in the generators have so changed the nature of the problem that knowledge based on earlier installations is not usually sufficiently profound to be applicable. Breadth of general experience, however, is giving brevity to the time required for meeting any specific set of conditions.

By maintaining maximum feed-water temperature as the water is heated in a well vented feed-water heater the generating system assures steam sufficiently free from association with oxygen and carbon dioxide to meet the requirements of the remainder of the system. By careful control and unceasing attention to the maintenance of proper chemical relations in the boiler water, the generating system assures uninterrupted supply of steam and minimum deterioration of piping and heating surfaces. By development work the generating system is approaching the solution of the problem of carry-over introduced by the pressure of economic development of steam generating units.

Problems of the Distributing System

By definition the distributing system is that part of the system where, uninterrupted by seasonal or operative conditions, any leakage is from the system into the surrounding atmosphere except only those infrequent times when shut downs are made for repairs or examinations.

Because any corrosive action occurs only after the steam condenses into water, it is necessary to understand thoroughly the character of the steam and the distribution of its components in any condensate that forms. The determination of the different components is accomplished with the use of the best laboratory methods available and in the case of oxygen is exact but in the cases of *pH* value and carbon dioxide the available methods do not permit the same order of accuracy as with oxygen. Because of the low capacity factor of pure water for dissolving iron oxygen can be considered as the capacity factor in corrosive action and the *pH* value as the velocity factor.

The Velocity Factor—pH Value and Carbon Dioxide

The *pH* value of entirely pure water is approximately 7 when measured at room temperature. The value actually obtained in careful distillation of water, as for instance in the laboratory still, ranges from about 5.5 to 6.5. Only by the most careful manipulation of such water following its condensation to provide certain removal of all gases such as carbon dioxide and sulphur dioxide, which are acid anhydrides, will the value of 7 be obtained (see Acree and Fawcett, Loc. cit.). Thus the *pH* values of Table 11, Section 4, taken in 1928, show that the condensate had acquired alkalinity from some source, and of course, most probably by carry-over of alkaline boiler water in the steam. The lower *pH* values of the other data of more recent months are therefore proof of lesser, if any, appreciable quantity of alkaline carry-over in the steam reaching the utilization system.

By definition, a *pH* value of 7 represents neutrality, above 7, alkalinity, and below acidity. As noted, distilled water usually lies on the acid side, as indicated by the *pH* value of 5.5 to 6.5, although it should be remembered that in an unbuffered water of this sort, the correct determination of *pH* value represents no simple task, and an error of several tenths or even a unit may readily be introduced into the determination.

The samples of Table 5, and those of Table 11 designated *steam* samples were obtained by condensing the steam in a cooling coil, particular attention being paid that all gases in the steam should dissolve as condensation occurred. The *pH* values of those samples range from a minimum of 4.7 to a maximum of 6, with a general average of 5.1. Excepting from Table 11, No. 2 of Section 1, regarding which some doubt exists; and Nos. 9 of Section 1 and Nos. 3 and 7 (second steam sample) of Section 2, in which the accumulation of carbon dioxide following partial condensation of the steam is plainly in evidence, the carbon-dioxide content of these samples range from a minimum of 13 ppm. to a maximum of 28.3 ppm. with a general average of 19.6 ppm.

The samples of Table 11 designated as *condensate,* were also passed through the cooling coil, but condensation had occurred in contact with steam as vapor phase and later contact had been made with the vapor phase present in the traps. The *pH* values of these samples (exclusive of Section 4 of Table 11) vary from 4.9 to 6.7, with general average of 5.8; the carbon-dioxide content varies from 0 to 22.3 ppm. with a general average of 7.6 ppm.

As the evaluation of these very small quantities of carbon dioxide is based upon the color change of an indicator, too great emphasis must not be placed on the numerical values. However, the content of carbon dioxide is markedly lower in the *condensate* than in the *steam* samples, as it should be to be in accord with the Law of Henry and Dalton. The higher *pH* values of the *condensate* samples show that the actual condensate which contacts the surfaces is of more desirable characteristics in this regard than would be considered the case if conclusions were based on the *steam* samples, as is frequently done.

The boiler water at Kips Bay is always maintained alkaline, and hence there is no possibility that acid anhydride other than carbon dioxide will be associated with its steam. This is the condition that obtains very generally in all steam derived from carefully conditioned boiler water. In the consideration, therefore, of any corrosive effects arising from *pH* value below neutrality, (for instance, 5.1 for condensed steam and 5.8 for condensate) and unassisted by any quantity factor, the discussion may be limited to the development due to carbon dioxide.

Conditions at the Internal Surfaces

Any corrosive action by the steam occurs only after its condensation into water. Since the rate factor of dissolution is dependent on the *pH* value of such condensate and since any *pH* value decrease below *7* is dependent in turn upon the amount of dissolved CO_2, it is necessary to find how much carbon dioxide can dissolve in condensate contacting the steam under consideration which contains 19.6 ppm. of carbon dioxide.

Carbon dioxide obeys the Law of Henry and Dalton

Table 10—Solubility of Oxygen and Carbon Dioxide at Different Partial Pressures in the Condensate from Steam at the Absolute Pressures of 165 Lb and 14.696 Lb[a]

Oxygen

1.	Steam pressure (lb absolute)	165	165	14.696	14.696
2.	Temperature (F)	366	366	212	212
3.	Vol. 1 lb water as saturated steam (cu ft)	2.748	2.748	26.82	26.82
4.	Oxygen in sample condensed with cooling coil (ml/liter)	0.3	1.0	0.3	1.0
5.	Solubility assumed at 15 lb partial pressure (ml/liter)	16	16	17	17
6.	Partial pressure of oxygen (lb)	0.000043	0.000144	0.0000034	0.0000115
7.	Solubility in the condensate (ml/liter)	0.000046	0.000153	0.0000039	0.000014

[a] Since these calculations are based on approximate data, the value 15 lb has been used as atmospheric pressure at the higher pressure used as an example.

Carbon Dioxide

1.	Steam pressure (lb absolute)	165	14.696
2.	Temperature (F)	366	212
3.	Vol. 1 lb water as saturated steam (cu ft)	2.748	26.82
4.	Carbon Dioxide in sample condensed with cooling coil (ppm)	19.6	19.6
5.	Solubility assumed at 15 lb partial pressure (ppm)	390	490
6.	Partial pressure of Carbon Dioxide (lb)	0.00147	0.00012
7.	Solubility in the condensate (ppm.)	0.038	0.004

Table 10*A*—Calculations for Table 10

Oxygen

—1—

(*a*) Steam Pressure—150 lb gage
Oxygen Content—corresponding to 0.3 ml/l. in a condensed sample drawn through cooling coil, thus retaining all oxygen.
At 150 lb gage pressure, 1 lb water = 2.748 cu ft saturated steam.
O_2 = 0.3 ml/l. = 0.42 ppm. = 0.00000042 lb per pound water.
At 0 deg C. and 760 mm 1 lb O_2 = 317000 ml. = 11.2 cu ft.

For 0.3 ml/l. = 0.00000042 lb O_2 per pound water.
0.00000042 × 1.71 = 0.00000072 cu ft.

Partial pressure of O_2 at 150 lb gage pressure

$$\frac{0.00000072 \times 165}{2.748} = 0.000043$$

At 150 lb gage pressure = 366 F = 185.6 C.

$$1 \text{ lb } O_2 = 11.2 \times \frac{273 + 185.6}{273} \times \frac{15}{165} = 1.71 \text{ cu ft.}$$

Solubility of O_2 in water at this partial pressure—
Assume solubility = 16 ml/l. at 185.6 C and 14.696 lb O_2.

$$\frac{16 \times 0.000043}{15} = 0.000046 \text{ ml/l.} = 0.000066 \text{ ppm.}$$

(*b*) O_2 = 1.0 ml/l. = 1.4 ppm. = 0.0000014 lb O_2 per pound water.
0.0000014 × 1.71 = 0.0000024 cu ft.
Partial pressure:

$$\frac{0.0000024 \times 165}{2.748} = 0.000144 \text{ lb.}$$

$$\frac{16 \times 0.000144}{15} = 0.000153 \text{ ml/l.} = 0.00022 \text{ ppm.}$$

—2—

(*a*) At 14.696 absolute pressure = 0 gage pressure
1 lb water = 26.82 cu ft saturated vapor
At 212 F = 100 deg C and 14.696 lb pressure

$$1 \text{ lb } O_2 = 11.2 \times \frac{373}{273} \times \frac{14.696}{14.696} = 15.3 \text{ cu ft.}$$

For 0.3 ml/l. = 0.00000042 lb O_2 per pound water
0.00000042 × 15.3 = 0.0000064 cu ft.
Partial pressure at 14.696 lb.

$$\frac{0.0000064 \times 14.696}{26.82} = 0.0000034 \text{ lb.}$$

Sol. O_2 = 17 ml/l. at 212 F and 14.696 lb O_2

$$\frac{17 \times .0000034}{14.696} = 0.0000039 \text{ ml/l.} = .000006 \text{ ppm.}$$

(*b*) For ml/l. of O_2, solubility is 0.000014 ml/l. = .00002 ppm.

Carbon Dioxide

—1—

Steam Pressure—150 lb gage
Carbon Dioxide—19.6 ppm.
At 150 lb gage pressure, 1 lb water = 2.748 cu ft saturated steam
CO_2 = 19.6 ppm. = .0000196 lb per pound water
At 0 deg C and 760 mm. 1 lb CO_2 = 231000 ml. = 8.2 cu ft.
At 150 lb gage pressure = 366 F = 185.6 deg C.

$$1 \text{ lb } CO_2 = 8.2 \times \frac{273 + 185.6}{273} \times \frac{15}{165} = 1.25 \text{ cu ft.}$$

For 19.6 ppm. = 0.0000196 lb CO_2 per pound water
0.0000196 × 125 = 0.0000245 cu ft.

Partial pressure of CO_2 at 150 lb gage pressure

$$\frac{0.0000245 \times 165}{2.748} = 0.00147 \text{ lb.}$$

Solubility of CO_2 in water at this partial pressure—
Assume sol. CO_2 = 390 ppm. at 185.6 deg C and 14.696 lb CO_2

$$\frac{390 \times 0.00147}{15} = 0.038 \text{ ppm.}$$

—2—

At 14.696 lb abs. pressure = 0 gage pressure
1 lb water = 26.82 cu ft sat. vapor.
At 212 F = 100 deg C and 14.696 lb pressure

$$1 \text{ lb } CO_2 = 8.2 \times \frac{373}{273} \times \frac{14.696}{14.696} = 11.2 \text{ cu ft.}$$

For 19.6 ppm. CO_2 = 0.0000196 lb per pound water
0.0000196 × 11.2 = 0.00022 cu ft
Partial pressure at 14.496 lb

$$\frac{0.00022 \times 14.696}{26.82} = 0.00012 \text{ lb}$$

Sol. CO_2 = 490 ppm. at 212 F and 14.696 lb CO_2

$$\frac{490 \times 0.00012}{14.696} = 0.004 \text{ ppm.}$$

under the conditions obtaining in the steam lines. Since the solubility of carbon dioxide is known (approximately) for the partial pressure of 14.696 lb (or one atmosphere) over a considerable temperature range, the amount of carbon dioxide in solution in condensate formed from and in contact with steam at any chosen pressure can be calculated, if the concentration of carbon dioxide in the steam is known. For example, assume that the laboratory test on a sample obtained by condensing steam of 165 lb absolute or 150 lb operating pressure in a cooling spiral gives the result of 19.6 ppm. of carbon dioxide. With each pound of steam, therefore, there is associated 0.0000196 lb of carbon dioxide. The volume of 1 lb of carbon dioxide at 366 F which corresponds to the operating pressure of 165 lb absolute is 1.25 cu ft. The volume of 0.0000196 lb of carbon dioxide for these conditions is $0.0000196 \times 1.25 = 0.0000245$ cu ft. The volume of 1 lb of water for the same conditions is 2.748 cu ft of saturated steam. Thus, 0.0000245 cu ft of carbon dioxide is associated with 2.748 cu ft of steam at absolute pressure of 165 lb, hence the partial pressure of carbon dioxide is

$$\frac{0.0000245 \times 165}{2.748} = 0.00147 \text{ lb.}$$

The solubility of carbon dioxide at the temperature of 366 F and a partial pressure of 14.696 lb is approximately 390 ppm. Hence at partial pressure of 0.00147 lb the solubility of carbon dioxide is

$$\frac{0.00147 \times 390}{15} = 0.038 \text{ ppm.}$$

These calculations and data are summarized in Table 10*A*. Calculations are also given for steam at atmospheric pressure. Thus with 19.6 ppm. of carbon dioxide present in the steam, any condensate in the 150-lb lines is saturated by 0.038 ppm. of dissolved carbon dioxide. In the lines with steam at atmospheric pressure, saturation of condensate occurs at 0.004 ppm. At intermediate pressures, the saturation values are intermediate of these values. The question of points at which the steam is stagnant and condensing, is considered in the discussion of oxygen.

The conclusion is obvious and is supported and emphasized by the data on condensate samples in Table 11. So long as the partial pressure of the carbon dioxide remains of the order of magnitude noted in line 7, Table 10, enough dissolution of the gas will not occur in the condensed steam to do any damage whatsoever. Doubling or trebling of the carbon dioxide content used as example, still would leave the partial pressure in this range. This relationship holds throughout the entire region of surfaces exposed uninterruptedly to the steam of pressure equal to atmospheric or greater.

Thus any deductions based directly on the *pH* values and carbon dioxide content of samples condensed with special attention to retention of the gases are inapplicable to the internal surfaces as defined in the preceding paragraph. They are applicable only at points of condensation, where said condensation occurs under circumstances akin to those surrounding the taking of samples.

pH Value in Absence of Oxygen

As noted heretofore in the section on results of simultaneous action of water on iron and carbon dioxide, in the absence of the quantity factor, oxygen, the tendency of the metal to dissolution is increased by lowered *pH* value, but so long as this lowering is effected by the dissolved carbon dioxide, the capacity of the water for dissolution of the metal is small because the dissolved iron itself readily builds the *pH* value to protective proportions. It is this fact in conjunction with the small partial pressure of the carbon dioxide, and therefore its slight solubility in the condensate, which has allowed the degree of carbon-dioxide removal to be a somewhat secondary consideration. The right thing to do, of course, is to restrict the carbon dioxide as much as economically possible, in order to prevent, so far as possible, any small dissolution, and also to avoid any danger of augmented rate and quantity of dissolution in case any concentration of oxygen is present for any reason.

The Quantity Factor—Dissolved Oxygen

The possibilities in deoxygenation of feed water consist of (1) use of deaerating heater, which if operating correctly, mechanically reduces the dissolved oxygen content to 0.025 to 0.05 ml. per liter; (2) use of open, direct contact heater, with temperature maintained at 212 F (if necessary, by means of live steam bled thereto) and with sufficient steam passing the vent to maintain low partial pressure of oxygen in the heater, in which case the dissolved oxygen content is reduced to 0.1 to 0.3 ml. per liter; (3) use of open, direct-contact heater, with no special precautions regarding maintenance of temperature or venting, in which case the dissolved-oxygen content of the effluent water may vary from 0.1 ml. upward, sometimes reaching 2 or 3 ml. per liter; (4) use of closed heater, in which case all oxygen dissolved in the feed water passes to the boiler and thence to the steam. In (1) and (2), a further step in removal of oxygen may be made by chemical fixation thereof following the heater, reducing it to zero if desired.

What degree of deoxygenation is essential for the production of steam that is entirely satisfactory for heating purposes? That question must be answered by the possibilities of corrosion that will ensue when the steam condenses, since corrosion of the system is not caused by the steam, but occurs only as the condensate simultaneously in contact with metal and gases, especially oxygen, makes possible their reaction with each other in dissolved form. Since the solubility of the gases under these conditions is a function of their partial pressure in the gaseous phase, the answer to this question devolves totally on these solubility relations.

The authors believe that the use of a closed heater represents introduction of unnecessarily large amounts of gases into the steam, and should be avoided. On the basis of the analysis that follows, any of the other three types of heaters provides essential limitation of oxygen. The preference of the authors is for (1) or (2), since either, when operating properly, acts as a governor providing a limiting concentration of oxygen in the system, so far as the generating system is its source. Thus in the direct-control open heater, with ample venting by pure steam, the oxygen in the effluent water should be zero theoretically, but in reality will be as a customary maximum, 0.3 ml/liter, and on the whole will be less than this figure. Even this value, however, represents super-saturation, and once the water is vaporized, thus releasing all dissolved oxygen, any condensation of steam that occurs will contain but an extremely small amount of oxygen when saturated at the original partial pressure of oxygen involved, and the tendency will be to undersaturation rather than saturation, and oversaturation will be an impossibility.

The pressure in this second division of the system may vary from a usual maximum of 150 lb gage to 0 lb gage. At no point in the system, therefore, will there be tendency for the extraneous atmospheric gases to gain entrance into the system. Hence, definite figures can be obtained on the quantity of dissolved oxygen that can be present at saturation in any condensate that forms in contact with the flowing steam at these various pressures, because oxygen obeys the Law of Henry and Dalton, and the partial pressure of oxygen corresponding to any amount thereof present in the effluent water from the heater, or found by tests on condensed samples of the steam, is readily determined.

In Table 10-*A* calculations similar to those for carbon dioxide are given for arriving at the amount of oxygen at saturation in condensate in contact with steam at 150 lb and 0 lb gage pressure respectively, and for an oxygen content in the steam of 0.3 and 1.0 ml/liter. In Table 10, line 7, are presented the saturation concentrations of oxygen under these various conditions. From line 6 it may be noted that the partial pressures of oxygen in the steam are extremely small, and hence in the condensate are found very small saturation values of less than 0.0002 ml/liter in any of the cases illustrated.

This quantity (0.0002 ml/liter) of dissolved oxygen is extremely small. It means, therefore, that for all those internal surfaces of the system in which the partial pressure of oxygen in the flowing steam is uninterruptedly the controlling factor defining the quantity of dissolved oxygen, the quantity factor in dissolution of the metal is quite negligible, even though determination of oxygen in the accepted manner by condensation of the steam should show a total of 1 or 2 or even more ml/liter.

It would seem that one is prone to misinterpret the significance of this determination of total oxygen—to magnify in one's thoughts some thousand fold the possible corroding capacity of this condensate, because subconsciously one assumes that the condensate in the system in contact with the steam is the same as the dissolved oxygen concentration that is found in the specially-taken sample for total oxygen determination. Thus far, the discussion has been carefully limited to flowing steam, and it has been pointed out that this is not the case since oxygen closely obeys the Law of Henry and Dalton. What will be true, however, where the steam is practically stagnant, and the condensate is discharged through a trap? In a radiator, for example?

Assume 0.3 ml/liter total oxygen in the incoming steam, and 0 lb gage pressure, *i. e.* atmospheric pressure on the radiator. Condensate is discharged periodically to a trap, but no gases are vented from the radiator. Uniformity of temperature (212 F) and of composition of vapor throughout the radiator is postulated.

When final equilibrium is reached, the condensate will contain 0.3 ml/liter. That amount will be its final maximum, because that is just the amount of oxygen furnished by the incoming steam.

When this occurs, the partial pressure of oxygen in the vapor phase in the radiator must be 0.265 lb or in other words, the vapor will be 1.76 per cent oxygen by volume, and since the original steam contains 0.000024 per cent, this value represents a concentration of more than 73000 fold. If one were to sample the vapor in the radiator as in the customary oxygen determination, taking care that no gaseous bubbles escape, it would be found that the oxygen concentration in the steam is 22,000 ml/liter.

Fig. 2 presents data on the temperatures and pressure in a radiator during operation. The radiator was of 20-in. height, 7 tubes, 20 sections, 73.33 sq ft area and 1.186 cu ft volume. The steam temperature *T* was measured by a mercury thermometer at top center of radiator; other temperatures *TC* were taken with thermocouples, inserted in sections 8 and 19, numbered from inlet end, and on the right side of one facing in direction of steam flow; and 2 and 14, on the left side. The pressure in the radiator was recorded in inches of mercury. The thermocouples were inserted 3 in. from the bottom of the radiator in second tube from edge.

The recorded steam temperature, once established, remained practically constant during the experiment, until the inlet and outlet valves of the radiator were closed, when it fell nearly uniformly with the other recorded temperatures. The temperatures as recorded by the thermocouples varied considerably at each position, and the temperatures of the several positions were different, and varied without any apparent relation to number of sections from inlet end.

The pressure was nearly constant until the inlet and outlet valves were closed, when it rapidly fell to —1.3 in. mercury, then over the next 45 min gradually rose to atmospheric pressure. This increase of pressure is very significant, as it denotes the readiness with which inleakage of air occurs. Had the system been tight, the pressure at the end, when the temperature was approximately 90 F would have been —28.5 in. of mercury plus whatever pressure there was from gases derived from the steam. The inleakage was more rapid, the larger the vacuum created by condensation of the steam, as shown by the decreasing slope of the pressure-time curve as atmospheric pressure was more nearly attained.

Condensation in the radiator may be assumed to be ¼ lb per hour per square foot of heating surface. Therefore condensation per hour was $\frac{1}{4} \times 73.33 = 18.33$ lb. Since each pound of water occupies 26.82 cu ft of space as saturated steam at 212 F, the 18.33 lb represents 492 cu ft of steam. Thus in the course of an hour, the condensation is $492 \div 1.186 = 415$ *radiatorsful.* In 24 hours therefore, 9,960 radiatorfuls of steam condense.

On the supposition that composition of vapor is uniform throughout the radiator, that condensate is discharged periodically to a trap, but no gases are vented from the radiator—a supposition that results in higher partial pressure of oxygen in the radiator than actually exists—and that the incoming steam contains 0.3 ml/liter of oxygen, the oxygen concentration in the radiator at the close of the period is 0.23 per cent (approximately).

Condensate at 212 F in contact with this vapor con-

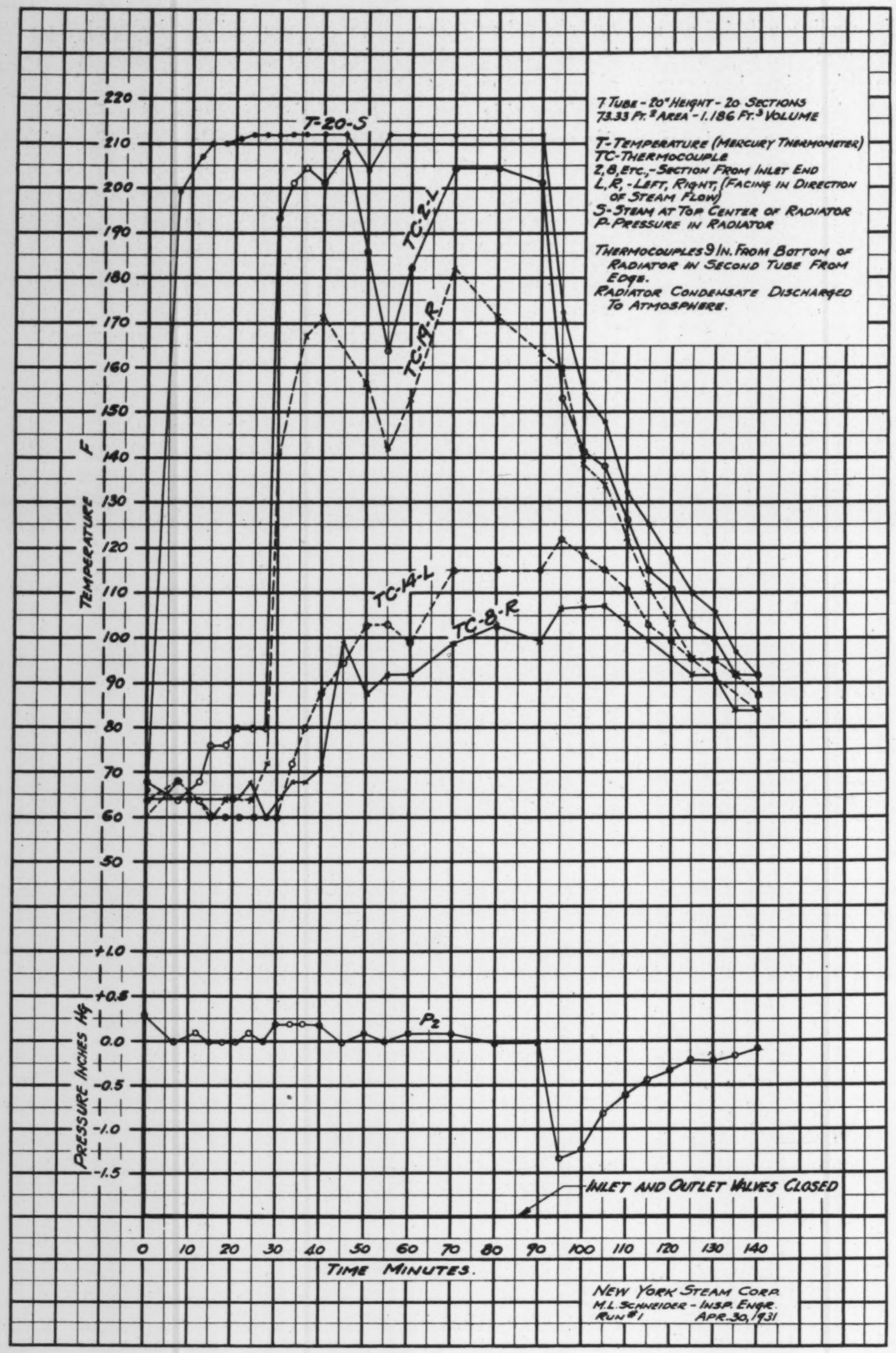

FIG. 2—DATA ON THE TEMPERATURES AND PRESSURES IN A RADIATOR DURING OPERATION

tains at saturation 0.23 (per cent) × 17 (ml/liter solubility of oxygen at 212 F and partial pressure of 14.696 lb) = 0.039 ml/liter. At the temperature of approximately 100 F recorded by *TC-14-L* and *TC-8-R*, at which the solubility of oxygen is 23 ml/liter, the saturation value is 0.053 ml/liter. If the steam contained 1 ml/liter, in place of the 0.3 ml/liter assumed in these calculations, the concentration of oxygen in the vapor at the close of a 24-hour period on the postulation of no loss during this period, would be approximately 0.77 per cent, and oxygen saturation in the condensate would be 0.13 ml/liter at 212 F and at 100 F, 0.18 ml/liter.

These figures are of interest because they demonstrate the relatively slight possible concentration of oxygen in the radiator, if its source is the steam. The low temperature of two of the thermocouples might of course be explained as due to the segregation of non-condensable gases (oxygen, nitrogen, carbon dioxide) in certain sections of the radiator, thus making possible a somewhat higher content of dissolved oxygen in the condensate having contact therewith. The fact that this occurs only in some sections, and that the temperature of condensate discharged at trap or sump is 140 — 170 F is proof that any such condition is not general nor permanent. Because of lack of sufficient turbulence in the confines of the radiator, the tendency is rather to obviate any concentration of gases by their removal. Thus the operation of the thermostatic valve for discharging condensate from the radiator is equivalent to blowdown on a boiler at the point of highest concentration of salts in the boiler water. The leak-valve on other radiators is comparable to continuous blowdown on the boiler. The biggest element in the oxygen-history of the radiator confines, however, is not the oxygen derived from the steam during the operating periods, but the composition of the vapors therein during the intermittent periods when the steam is turned off. As shown by the curves of Fig. 2, the vacuum developed in the radiator when the steam supply was shut off, was rapidly dissipated denoting ingress of air to take the place of the condensing steam. When the steam is again turned on, heating occurs first near the steam-entry, and gradually extends through the radiator, while meanwhile the atmospheric gases are driven before the steam, and discharge practically completely through the open trap or leak-valve. During the off-periods, the concentration of oxygen in the radiator bears no relationship to the oxygen in the steam, but is a function of the composition of the atmosphere.

It seems certain, therefore, that the maximum concentrations of oxygen in the condensate, as calculated above, are much higher than those that actually exist, and that even with a ml/liter of oxygen continuously present in the steam going to the radiator, the amount dissolved in the condensate from this source is too small to be of detriment to the surfaces the condensate contacts.

It is apparent that it is not a simple matter to obtain a condensate containing a few tenths ml/liter of oxygen if the source of supply of that oxygen is steam likewise containing only a few tenths ml/liter. There is a possibility that this might occur at totally dead ends, devoid of any periodic flow, and unvented. As for samples taken elsewhere, as from traps in any of the working parts of the system, any justification for ascribing several tenths ml/liter of oxygen therein to the presence of a similar total oxygen content in the flowing steam seems utterly impossible and absurd. The same considerations apply as well to carbon dioxide as to oxygen.

Discussion of Data from Various Buildings

In Table 5 are presented data on station steam in the distributing mains serving the buildings under consideration. Table 11 presents further data from different points. As an average value of oxygen in the steam as it leaves the distributing mains of the generating system to enter the lines of the utilizing system, 0.5 ml/liter is generously high.

As shown in the preceding discussion, the partial pressure of oxygen in the steam, corresponding to this concentration of oxygen therein, is so low that the amount which can dissolve in any condensate is too slight to be considered a serious quantity factor in any corrosive action.

Discussion of returns before vacuum pumps is deferred to Division 3. At this point, however, it is pertinent to discuss the high oxygen values found in samples taken before the trap on the return from the hot water tank. These may be noted in Table 11 as follows: Sect. 1, Building *A,* Nos. 5 and 6; Sect. 2, Building *B,* No. 8; Sect. 4, Building *D,* Nos. 1 and 2.

In Building *A, steam* at the 5-lb header on the 29th

TABLE 11—OXYGEN AND CARBON DIOXIDE IN THE STEAM—SECTION 1.

Analyses of Steam or Condensate Drawn from Different Parts of the Steam System in Building A

(For conditions in the main steam lines on same dates, see Table 5)

No.	DATE 1930	WHERE TAKEN	STEAM OR CONDENSATE	OXYGEN ML/L	CARBON DIOXIDE, PPM.	pH VALUES
1	Aug. 5	St. John *C* Meter	Steam	0.39	20.0	5.5
2	Aug. 6	Downstream side of valve 8 in. H.P. riser from 90-lb. header	Steam	0.25	8.6	5.2
		St. John *C* Meter	Steam	0.34	0.0	5.2
3	Aug. 7	Trap at west end of 90-lb. header	Steam	0.61	13.0	5.1
4	Aug. 8	St. John *C* Meter	Steam	0.39	18.5	5.2
		Steam Trap at *C* Meter	Steam	0.46	20.0	5.5
5	Aug. 12	Before trap on return from hot water tank 29th Floor	Steam probably mixed with considerable condensate	1.12	25.0	4.9
6	Aug. 13	5-lb. header 29th floor	Steam	0.42	19.5	5.1
		Outlet of hot water tank trap, 29th floor	Condensate	1.10	6.0	5.8
7	Aug. 15	Return from trap on Coffee Urn No. 3 Kitchen Uptown Club	Condensate	0.35	22.3	5.1
8	Aug. 25	Inlet side of trap at heel of 8-in.	Steam	0.19	23.0	5.4
		Returns before vacuum pump	Condensate	4.48	2.0	6.6
9	Aug. 26	Returns on Pressure Gage System	Steam	0.7	39.0	5.8
10	April 21 Temp. of Condensate 56F.	St. John's Meter		0.35[a]	15.2[b]	5.3
11	April 21 Temp. of Condensate 60F.	19th floor		0.26[c]	16.4[d]	5.3

[a] Avg. of 6 tests, varying from 0.26 to 0.43 ml/l.
[b] Avg. of 5 tests, varying from 13.1 to 16.9 ppm.
[c] Avg. of 4 tests, varying from 0.21 to 0.34 ml/l.
[d] Avg. of 4 tests, varying from 15.3 to 16.4 ppm.

floor contained 0.42 ml/liter of oxygen, 19.5 ppm. of carbon dioxide; *steam* before the trap on the return from the hot water tank contained 1.12 ml/liter of oxygen and 25 ppm. of carbon dioxide. On the other hand, the *condensate* from the outlet to the hot water tank trap contained 1.10 ml/liter of oxygen and 6.0 ppm. of carbon dioxide. The correctness of the analytical work in showing less carbon-dioxide content in the condensate is well illustrated by the *pH* values: When steam was condensed for sampling, using cooling coil, and getting into solution the 19.5 and 25 ppm. of carbon dioxide, the *pH* values were respectively 5.1 and 4.9; on the condensate, containing only 6.0 ppm. of carbon dioxide, the *pH* value was 5.8.

The steam in the hot water tank showed an increase in concentration of oxygen over that in the steam in the header of 2.7 fold; the carbon dioxide increase was 1.3 fold. This decrease in relative concentration of carbon dioxide leads to the suspicion that the increment of oxygen concentration was due in part to inleakage of air in which the ratio of oxygen to carbon dioxide is 500 : 1, whereas in the steam the ratio is 1 : 32.5. Suspicion changes to certainty when the dissolved gases in the condensate are considered. By the widest stretch of

TABLE 11—OXYGEN AND CARBON DIOXIDE IN THE STEAM—SECTION 2

Analyses of Steam or Condensate Drawn from Different Parts of the Steam System in Building B.

No.	DATE 1930	WHERE TAKEN	STEAM OR CONDENSATE	OXYGEN ML/L.	CARBON DIOXIDE PPM.	*pH* VALUES
1	Nov. 23	Condensate E. Meter	Steam	0.43	25.9	5.1
2	Nov. 26	High Pressure drip from valet shop	Condensate	0.29	5.0	5.6
3	Nov. 25	Valet shop condensate at sub. meter	Steam	0.53	34.1	5.0
4	Nov. 25	Trap-outlet of presser valet shop	Condensate	0.27	3.0	5.3
		L.P. trap valet shop	Condensate below atmospheric press	1.43	5.2	5.1
5	Dec. 11	Grill room kitchen entering steam table	Steam	0.99	23.9	5.0
6	Dec. 3	Laundry—condensate from entrance to presser	Steam	0.75	22.9	5.0
		Laundry—condensate from entrance to mangle	Steam	0.49	24.0	5.0
7	Nov. 25	Laundry returns from presser before trap	Steam	0.58	28.3	4.9
		Laundry returns from mangle after trap	Condensate	1.50	20.1	4.9
		Laundry returns from mangle before trap	Steam	2.58	52.5	4.8
		Laundry return—basement—before pre-heater	Condensate	0.20	2.1	6.0
8	Dec. 1	Trap-inlet to hot water tank	Condensate	3.99	10.3	5.6
		Before trap inlet to hot water heater	Steam	1.10	15.8	5.1
9	Jan. 9, '31	Kitchen returns—after trap before manifold	Condensate	2.93	19.3	5.3
		Same. (See Notes.)	Condensate	1.21	10.6	5.6
10	Jan. 9, '31	Drip lines, H. P. risers before trap and manifold, under pressure. Sampled on left side	Condensate	0.79	None found	6.7
		Same. Sampled on right side, under pressure	Condensate	0.90	None found	6.7
		Attempt to take same during vacuum period	Condensate	3.40	None found	6.7

TABLE 11A

NOTES ON TABLE 11—SECTION 2

1. *Condensate E Meter.* Values are average of four determinations in each case.
2. *High Pressure Drip from Valet Shop.* Sample taken in subbasement as 4th floor location was inaccessible. Values are average of determinations in each case.
3. *Valet Shop Condensate at Sub. Meter.* Sample could not be taken from trap at meter. Taken direct from meter. Values are average of three determinations in each case.
4. *Trap—Outlet of Presser Valet Shop.* Average of three determinations in each case.

   L. P. Trap, Valet Shop. Average of four determinations in each case.
5. *Grill Room Kitchen. Entering Steam Table.* Only permissible location in Grill room. Values are average of seven determinations on oxygen, six each on carbon dioxide and *pH* value.
6. *Laundry—Condensate from Entrance to Presser.* Average of three determinations in each case.

   Laundry—Condensate from Entrance to Mangle. Average of four determinations on oxygen, and three each on carbon dioxide and *pH value.*
7. *Laundry Return from Presser before Trap.* Average of four determinations in each case.

   Laundry Return from Mangle After Trap. Impossible to eliminate bubbles. Average of four determinations in each case.

   Laundry Return from Mangle Before Trap. Impossible to eliminate bubbles. Average of five determinations on oxygen and four each on carbon dioxide and *pH* value.

   Laundry Return—Basement—Before Preheater. Average of three determinations each.
8. *Trap—Inlet to Hot Water Heater.* Average of six determinations in each case.

   Before Trap—Inlet to Hot Water Heater. Average of four determinations in each case.
9. *Kitchen Returns—After Traps Before Manifold.* Vacuum pump shut off, also line to small preheater and radiator bypass. Average of two determinations in each case.

   Same. With above valves open. Water was rusty. Average of two determinations in each case.
10. *Drip Lines, H. P. Risers Before Trap and Manifold, Under Pressure, Samples on Left Side.* These are not hot water tank returns. *NOTE:* When steam is on, there is pressure in these lines, when steam is off, there is vacuum. This analysis represents the pressure period. Average of two determinations in each case.

    Same. Sampled on Right Side Under Pressure. Average of two determinations in each case.

    Attempt to take same during Vacuum Period. Flow discontinuous, 4 in. mercury vacuum. One sample each for oxygen and carbon dioxide, two for *pH* value.

TABLE 11—OXYGEN AND CARBON DIOXIDE IN THE STEAM
SECTION 3
Building C—(Report by New York Testing Laboratories)

No.	Date 1930	Where Taken	Oxygen ml/l.	Carbon Dioxide ppm.	pH Values
1	About April				
	Thursday	Incoming Steam	2.0[e]	18	5.5
	Friday	Incoming Steam	0.42[e]	15	......
	Saturday	Incoming Steam	0.30	18	......
	Monday	Incoming Steam	0.34	......	......
	50-lb. High Pressure Line to Kitchen				
2	Friday	Direct Return	Nil	37	......
		Return Drip	0.05	5	......
	Saturday	Roof Garden Kitchen on Pressure Side	Nil	......	......
	Low Pressure Returns				
3	Friday	Pump Running	4.9	0.5	......
	Saturday	Pump Running	4.1	......	......
		Pump Not Running	3.1	......	......

[e] Average of three tests.

TABLE 11—OXYGEN AND CARBON DIOXIDE IN THE STEAM
SECTION 4
Building D (Hall Laboratorics, Inc., Report)

No.	Date 1930	Where Taken	Oxygen ml/l.	Carbon Dioxide ppm.	pH Values
1	About January				
		North Heater	3.5	......	......
		before trap	3.2	......	......
		North Heater	1.4	......	......
		after trap	1.0	......	......
2		South Heater	......	......	......
		before trap	0.8	......	......
		South Heater	1.7	......	......
		after trap	0.5	......	......
3		St. John Meter	0.5[f]	......	......
4		At Kips Bay	0.9	......	......
5		Condensate Receiving Tank	......	......	7.0
6		Steam Corporation trap at inlet	......	......	8.2
7		Hot water Boiler Trap	......	......	7.3
8		Laundry Trap on Mangle	......	......	8.4
9		No. 6 Heating Meter	......	......	7.4

[f] Average of four tests varying from 0.2 to 0.7 ml/l.

the imagination, steam containing 1.12 ml/liter of oxygen cannot form condensate at 140 — 170 F containing 1.10 ml/liter. On the basis of the Law of Henry and Dalton, saturation concentration is only 0.000016 ml/liter. Hence immense enrichment in oxygen must have occurred in the vapor phase contacting the sample taken from the trap.

Just how the condensate picks up its dissolved carbon dioxide is hard to explain. However, as noted heretofore, while the values of the carbon-dioxide determinations relative to each other are exact enough, the methods do not permit attaching too great significance to the absolute values.

Thus it is quite evident that all of the water coming to the traps from the hot water tank cannot contain one or more ml/liter of dissolved oxygen, if the source of that oxygen is the steam itself, because there is not that much oxygen associated with the steam. A part of the water might have this much oxygen, and the rest be relatively free from it. This interpretation of conditions, however, is negated by the fact that the samples taken from the outlet of the hot water tank trap show an amount of dissolved oxygen that is higher or as high as that in the steam. Doubtless, therefore, this larger supply of oxygen is obtained mainly not from the steam but from the inexhaustible source of the extraneous atmosphere.

Other examples that well illustrate these facts are found in Table 11, Section 1, Building *A,* No. 8; Section 2, Building *B,* Nos. 3 and 4, and Nos. 6 and 7; Section 4, Building *D,* Nos. 1 and 2, in which the samples before the trap were steam, those following the trap were condensate.

In the returns from the trap on the coffee urn No. 3 (kitchen) of the Uptown Club, Sect. 1, Building *A,* No. 7, only 0.35 ml/liter of oxygen is present. This fact calls attention to one factor in these pressure lines that must be remembered. At times it occurs that a shut-off valve on one of the pressure lines is located considerably back of the point at which the end of the pressure system ends, and the vacuum system begins. When this line is shut off, as in intermittent operation, then that section of the line between the shut-off valve and the normal beginning of the vacuum system in reality constitutes a part of the vacuum system and must be so considered in problems of corrosion.

In conclusion of the discussion on this division, attention is again directed to the boundaries set for it, which limits it to that part of the system, whether the distribution mains of the generating system, or the lines of the utilizing system, in which the steam pressure is always equal to or greater than the pressure of the surrounding atmosphere, and in which therefore flow of any leakage is from conduit into the atmosphere. There exists in this division no intermittency of steam pressure resulting from operating practices. Under these conditions, if the amount of oxygen and carbon dioxide associated with the steam as it leaves the generating system is limited to that characteristic of the effluent water from a direct contact vented heater pegged at 212 F, the partial pressure of these gases in the steam cannot possibly be sufficient to provide a concentration of either gas in any water condensed from the steam to provide either a deleterious quantity or rate factor of corrosion by said condensed water. In fact, the deductions from the Law of Henry and Dalton concede much greater leeway of oxygen content in steam than this without indicating it as causative of detrimental corrosive conditions in the utilizing system.

To Be Concluded
in December.

Importance of Radiation in Heat Transfer Through Air Spaces

By E. R. Queer[1], State College, Pa.
NON-MEMBER

HEAT is transmitted across confined air spaces by radiation, convection and conduction. It has been customary to regard each of these modes as an independent method but in this investigation it was found impractical to separate conduction and convection. These two modes may be conveniently designated by the term diffusion. Justification for this is based on the following conception of heat transfer in an air space:

Part of the heat is transferred by radiation in accordance with the familiar Stefan-Boltzmann law; the remainer is transferred by conduction and convection which are inextricably combined. Heat leaves the hot surface by conduction through the surface film, then it is carried across the air space by a diffusion process which is a combination of conduction and convection to the receiving surface where it again passes through the surface film by pure conduction.

The factors affecting radiation in confined air spaces are the emissivity of the surfaces and their temperatures; those affecting diffusion are height and width of the spaces and the conductivity of the enclosed fluid. This paper summarizes the results of a comprehensive series of tests carried on at the Engineering Experiment Station of The Pennsylvania State College in which these factors were varied to determine their relative effect on heat transmission.

Although considerable work has been performed and reported upon heat transfer through air spaces, yet the importance of radiant heat transfer is still disregarded by many engineers. Additional data are herein presented to stress the importance of radiant heat transfer, and factors affecting conduction and convection in thin air spaces.

The electric hot plate method[2] was used to measure the thermal resistance of the air spaces tested. The hot plate was placed between two similar test specimens and the heat transmitted was absorbed by cold water plates.

The effect of the size of air spaces was determined by using spaces varying in width from 1/3 in. to 1½ in. and in height from 4½ in. to 35 in. The effect of the nature of the surface on the radiation was determined by using polished aluminum and aluminum covered with lamp black which has an emissivity about the same as ordinary building materials.

Both the hot plate and the cold plates were covered with aluminum foil (0.003 in. thick) cemented on with a thin film of raw rubber. The aluminum foil gave an excellent bright surface that did not change appreciably with time. Black radiating surfaces were obtained by painting the aluminum foil with lampblack mixed with alcohol. This coating could be rubbed off easily leaving the original polished aluminum surface unmarred.

Temperatures were measured with single thermocouples cemented to the surfaces with a waterproof cement. A comparison between this method of attaching thermocouples and soldering them in a groove showed no appreciable difference in readings. The cement apparently forms a translucent covering that does not affect the emissivity over the couple; this is an important consideration in the measurement of the surface temperatures.

Emissivity and Radiation

The effect of various kinds of surfaces on the heat transmission through air spaces is shown by the data in Table 1 which are plotted in Fig. 1. Curves 1, 2 and 3 furnish striking evidence of the efficacy of bright metallic surfaces in reducing heat transmission by reducing the radiation. Curve 4 shows the effect of placing one bright metal screen between two black surfaces. In this case there are two air spaces each one-half the width of the original space; this reduced somewhat the diffusional heat transfer by conduction and convection as well as the radiation.

To separate the total heat transfer into its component parts the Stefan-Boltzmann law was used to calculate the part transferred by radiation. The following equation gives the law in the form used:

$$H_r = \frac{0.172 \times \dfrac{e_1\, e_2}{e_1 + e_2 - e_1\, e_2}\left[\left(\dfrac{T_1}{100}\right)^4 - \left(\dfrac{T_2}{100}\right)^4\right]}{n + 1} \qquad (1)$$

where

H_r = radiant heat transfer in Btu per hour per square foot
e_1 = emissivity factor for hot surface
e_2 = emissivity factor for cold surface
T_1 = absolute temperature of hot surface
T_2 = absolute temperature of cold surface
n = number of screens

$$E = \frac{e_1\, e_2}{e_1 + e_2 - e_1\, e_2} = \frac{e^2}{2e - e^2} \text{ (hot and cold surfaces alike)}$$

In the first series of experiments the combined emissivity factor E was determined. The bright aluminum surfaces of the heaters and cold plates were separated by one-inch air spaces as shown in Fig. 2. Two hot plates were placed one above the other in a horizontal position with a heat flow meter between them. Heat was supplied to each air space by its respective heater. The heat input to the lower plate was held constant, while that to the upper plate was varied to keep the

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[2] See Bulletin No. 37, Heat Flow Meters and Thermal Conductivity Measurements, by F. G. Hechler and E. R. Queer (Engineering Experiment Station, The Pennsylvania State College.)

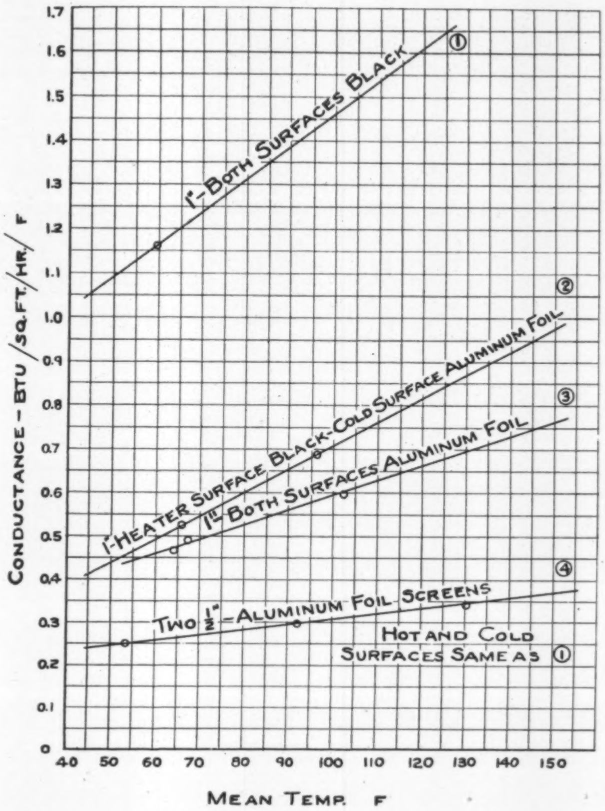

Fig. 1—Effect of Various Kinds of Surfaces on Heat Transmission Through Air Spaces

Table 1—Effect of Various Kinds of Surfaces on Heat Transmission Through Air Spaces (See Fig. 1)

Curve No.	Heat Input (Btu per hr per sq ft)	High Temp. (Deg Fahr)	Low Temp. (Deg Fahr)	ΔT (Deg Fahr)	MT (Deg Fahr)	C	Remarks
1	17.59	67.2	52.1	15.1	59.7	1.163	1 in. air space,
1	55.5	123.7	88.6	35.1	106.2	1.581	9 in. high, black surfaces
2	17.64	83.1	49.3	33.8	66.2	0.522	1 in. air space, cold surface, alfoil,
2	39.7	125.5	68.0	57.5	96.7	0.690	9 in. high, hot surface black
3	17.59	86.1	50.4	35.7	68.3	0.492	1 in. air space,
3	39.50	135.7	69.1	66.6	102.4	0.594	9 in. high, hot and cold surface, alfoil
4	9.90	84.3	44.8	39.5	64.5	0.251	two ½-in. air spaces,
4	22.23	129.5	55.0	74.5	92.3	0.299	9 in. high, aluminum screen
4	40.25	189.6	72.1	117.5	130.8	0.342	hot and cold surface black

temperatures of the two heater surfaces facing each other equal. (This equality was indicated by a zero deflection of the galvanometer connected across the heat flow meter). Hence all the heat of the lower heater test area was forced downward through the one-inch air space. By this arrangement convection was eliminated in this space, leaving only pure conduction and radiation. Deducting the heat transferred by conduction from the total heat input, the remainder was radiation. Substituting this heat transfer and the surface temperatures in Equation 1, E was calculated. A sample calculation follows:

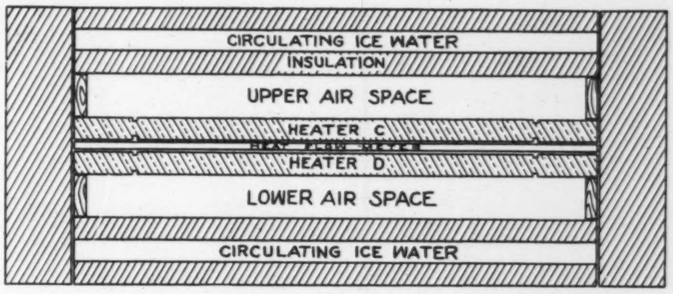

Fig. 2—Apparatus Used to Determine e, Showing Two Hot Plates with Heat Flow Meter Between

Hot surface temperature = 155.4 F + 460 = 615.4 F, absolute
Cold surface temperature = 53.8 F + 460 = 513.8 F, absolute
Temperature difference $\Delta T = 101.6$ F
Mean temperature (MT) = 104.6 F
Heat transferred by conduction through air
$= 0.172^* \times 101.6$
$\doteq 17.49$ Btu per hour per square foot
Total heat input = 26.15 Btu per hour per square foot
$H_r = 26.15 - 17.49 = 8.66$ Btu per hour per square foot
Hot and cold surfaces are alike, therefore $e_1 = e_2$

$$E = \frac{e^2}{2e - e^2}$$

$$= \frac{H_r}{0.172\left[\left(\frac{T_1}{100}\right) - \left(\frac{T_2}{100}\right)\right]^4}$$

$$= \frac{8.66}{0.172\left[\left(\frac{615.4}{100}\right)^4 - \left(\frac{513.8}{100}\right)^4\right]}$$

$= 0.0684$
$e = 0.128$ for these aluminum surfaces. These combined emissivity factors are tabulated in Table 2 and plotted as Curve 2, Fig. 3.

Effect of Spacing

A second series of experiments was made to show how the conductance of vertical air spaces varied with width

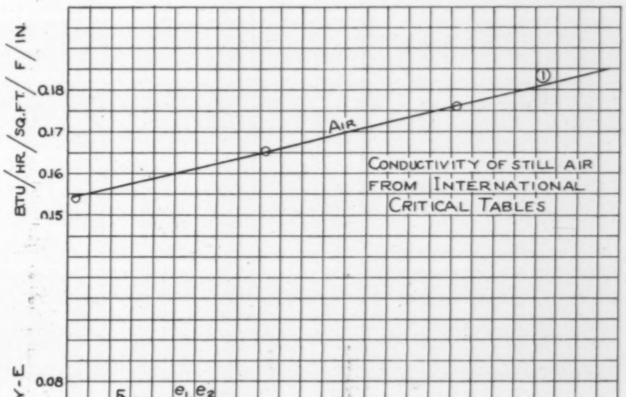

Fig. 3—Conductivities of Still Air and Combined Emissivity Factors

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*See Curve 1, Fig. 3. Conductivity of still air taken from the International Critical Tables.

TABLE 2—EMISSIVITY FACTORS

Heat Input (Btu per hr per sq ft)	High Temp. (Deg Fahr)	Low Temp. (Deg Fahr)	ΔT (Deg Fahr)	MT (Deg Fahr)	Heat Conduction (Btu per hr per sq ft)	H_r (Btu per hr per sq ft)	E	e*
16.65	116.0	46.8	69.2	81.4	11.57	5.08	0.0670	0.1256
26.15	155.4	53.8	101.6	104.6	17.49	8.66	0.0684	0.1283
44.53	220.7	67.5	153.2	144.1	27.90	16.63	0.0703	0.1310

*Aluminum used.

of the space. These results are plotted on Fig. 4. Two mean temperatures were chosen to indicate that the shape of the curve did not change over the range tested. It should also be noted that changing the height of the air space did not alter the shape.

An examination of Curves 1 and 5 of Fig. 4 reveals an optimum single space conductance at ¾ in. spacing. This is in agreement with the results of other investigators. Curves 2 and 4 on this same figure show the effect of spacing on horizontal air spaces. These data represent an average conductance of equal spaces above and below the heater. In the upper space, convection is at a maximum whereas it is eliminated in the lower one. This is equivalent to considering the combined loss from the top and bottom of an insulated box, the insulation being made up of a series of bright surfaces. No optimum spacing for the single horizontal space was reached in the range tested.

Effect of Height

Contrary to the general belief, it has been found that blocking thin air spaces decreases the insulating value of the space.[3] To show the magnitude of this effect a third series of experiments was made keeping the air space widths constant and varying the height from 4½ in. to 35 in. Tests at two mean temperatures were made to establish the position of a mean temperature-conductance curve. Fig. 5 shows a plot of some of these data for a one-inch space, and heights of 4½, 9, and 11⅝ in. The temperatures are also shown in this figure.

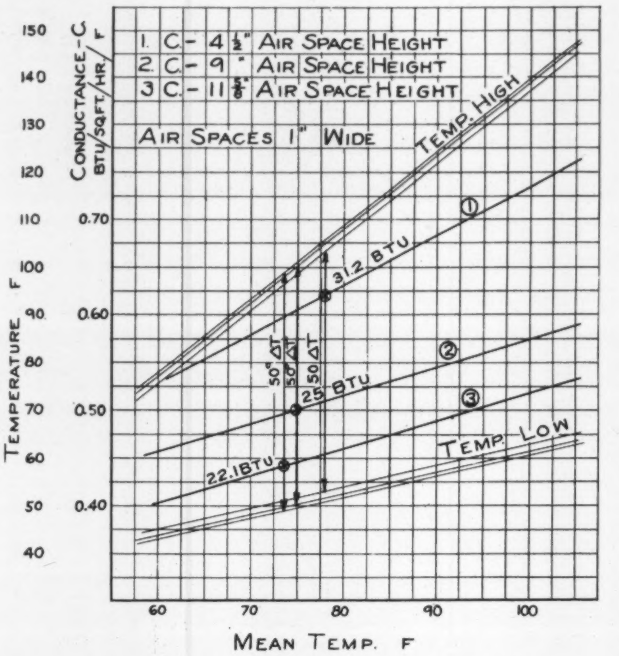

FIG. 5—MEAN TEMPERATURE—CONDUCTANCE CURVES

If the temperature difference at any mean temperature is multiplied by the corresponding conductance for the same mean temperature the product is the total heat flow. The radiation may be calculated from the corresponding temperatures of the hot and cold surfaces and the combined emissivity factor. Subtracting this from the total heat input gives the diffusional transfer. Repeating this for each height and width of air space gave the values for the heat transferred by conduction and convection as plotted in Fig. 6. The following example illustrates the method of calculation:

Example. A temperature drop between the hot and cold surfaces of 50 F was arbitrarily chosen. Height of air space, 9 in. Conductance $(C) = 0.50$ at a differential of 50 deg, MT $= 75$ F.

H (total) $= 0.50 \times 50 = 25$ Btu per hour per square foot.

High side temperature $= 100$ F $+ 460 = 560$ F, absolute

Low side temperature $= 50$ F $+ 460 = 510$ F, absolute

E at 75 F (MT) $= 0.0665$ (From Fig. 3, Curve 2)

$$H_r = \frac{0.172 \times 0.0665 \left[\left(\frac{560}{100}\right)^4 + \left(\frac{510}{100}\right)^4\right]}{1}$$

$= 3.54$ Btu per hour per square foot

H (diffusional) $= 25 - 3.54 = 21.46$ Btu per square foot per hour

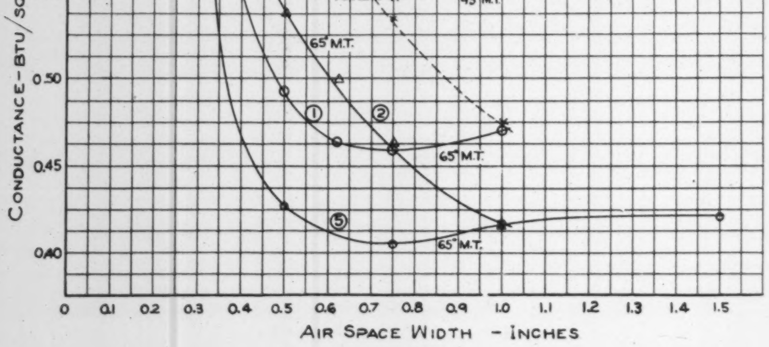

FIG. 4—VARIATION OF CONDUCTANCE OF VERTICAL AIR SPACES WITH WIDTH OF SPACES

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[3] See Testing Thermal Insulators, by H. C. Dickinson, M. S. Van Dusen (*A. S. R. E.*, Dec. 1915).

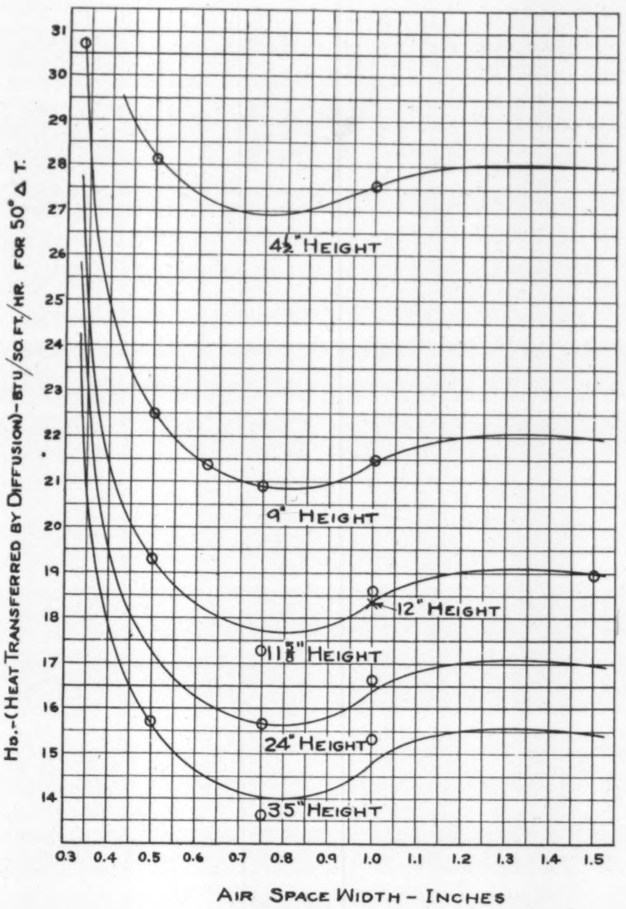

FIG. 6—CURVES SHOWING RELATION BETWEEN HEAT DIFFUSION AND AIR SPACE WIDTH

In like manner the other values of Fig. 6 were calculated. This figure is used merely to illustrate what will happen to the diffusional heat transfer at any temperature difference, while the height and width of spaces are changed.

This was a surprising phenomenon, since it has been the usual assumption that blocking thin air spaces decreased the conductance. The blocking was thought to decrease the fluid pressure differential between the hot and cold surfaces, consequently the diffusional heat transfer was decreased and the resistance to heat flow increased. Tests show that the reverse happened. Apparently what took place was that increasing the air space height increased the surface frictional resistance to air flow by an amount greater than the fluid pressure differential caused by differences in densities on the hot and cold surfaces. Hence, convection would be decreased and the resistance to heat flow increased. The curve of Fig. 7 is a cross plot of Fig. 6 at one-inch spacing. The shape of this curve would be the same for any other spacing.

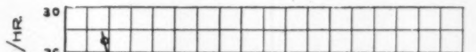

FIG. 7—CROSS PLOT OF FIG. 6 FOR ONE-INCH AIR SPACE

Summary

It has been generally supposed that most of the heat lost through air spaces of building structures was by conduction and convection. However, this is not the case; most of the heat transferred between conventional building materials is by radiation.[4] Several examples are cited to show the magnitude of the radiant heat transfer between surfaces of the type mentioned.

Wall[5] *No.* 23
$\Delta T = 16.7$ F for air space between 2×4 studs
$T_2 = 49.8$ F $+ 460 = 509.8$ F, absolute
$T_4 = 33.1$ F $+ 460 = 493.1$ F, absolute

$$H_r = \frac{0.172 \dfrac{0.93^2}{0.93 + 0.93 - 0.93^2} \left[\left(\dfrac{509.8}{100} \right)^4 - \left(\dfrac{493.1}{100} \right)^4 \right]}{1}$$

$= 12.65$ Btu per hour per square foot
Per cent of total heat transferred by radiation =

$$\frac{12.65 \times 100}{16.41} = 77 \text{ per cent}$$

If the 4-in. air space were divided into two 2-in. spaces by a thin bright metallic screen having an emissivity value 0.1, then

$$E = \frac{0.1 \times 0.93}{0.1 + 0.93 - (0.1 \times 0.93)} = 0.099$$

And if the same temperatures existed on the high and low surfaces

$$H_r = \frac{0.172 \times 0.099 \left[\left(\dfrac{509.8}{100} \right)^4 - \left(\dfrac{493.1}{100} \right)^4 \right]}{2} = 0.72 \text{ Btu}$$

per hour per square foot compared with a value of 12.65 Btu per hour per square foot for the original wall.

This same fact was borne out by the first series of experiments shown in Fig. 1, and accounts for most of the enormous reduction in the value of C between Curves 1 and 4.

Likewise for[5] Walls No. 25 and No. 30 respectively:
Per cent of total heat transferred by radiation =

$$\frac{11.32 \times 100}{15.10} = 75 \text{ per cent}$$

Per cent of total heat transferred by radiation =

$$\frac{8.86 \times 100}{12.3} = 72 \text{ per cent}$$

This analysis indicates that approximately 75 per cent of the total heat transfer is by radiation. An optimum insulation spacing for multiple bright metallic surfaces that are very thin has been found to be 0.37 in. That is, if one has a given space to be insulated, the maximum insulating effect for a minimum number of bright metallic sheets is obtained by spacing sheets at 0.37 in. By choosing any space width and dividing it into spacings varying from about ⅓ in. to ¾ in., with bright sheets of aluminum one can calculate the overall transmission factor for each spacing. Fig. 8 shows how this factor

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[4] See discussion by F. G. Hechler of R. H. Heilman's paper on Surface Heat Transmission. (*A. S. M. E.* Transactions—Fuels and Steam Power Division, Sept.-Dec. 1929. Vol. 51, No. 22, p. 299.)

[5] These data are taken from Heat Transmission Research, by F. B. Rowley, F. M. Morris and A. B. Algren (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928).

TABLE 3—SOME EXPERIMENTAL DATA

TEST No.	No. OF SPACES	SPACE WIDTH (IN.)	AIR SPACE HEIGHT (IN.)	HEAT INPUT (BTU PER HR PER SQ FT)	HIGH TEMP. SIDE (DEG FAHR)	LOW TEMP. SIDE (DEG FAHR)	TEMP. DIFF. ΔT (DEG FAHR)	MT (DEG FAHR)	C (BTU PER HR PER SQ FT)
1	1	½	9	17.58	83.0	47.2	35.8	65.1	0.491
2	1	½	9	39.55	130.2	60.9	69.3	95.5	0.571
3	1	½	11⅝	17.58	84.1	47.3	36.8	65.7	0.478
4	1	½	11⅝	39.55	134.7	62.3	72.4	98.5	0.546
5	2	½	9	17.58	116.8	49.0	67.8	82.9	0.259
6	2	½	9	39.55	195.7	63.8	131.9	129.8	0.300
7	2	½	11⅝	17.58	122.2	48.7	73.5	85.4	0.239
8	2	½	11⅝	39.55	214.0	64.3	149.7	139.6	0.264
9	1	⅓	9	17.58	75.9	47.9	28.0	61.9	0.627
10	1	⅓	9	39.55	117.9	61.9	56.0	89.9	0.706
11	3	⅓	9	9.9	88.1	44.2	43.9	66.2	0.225
12	3	⅓	9	22.25	145.0	55.0	90.0	100.0	0.247
13	3	⅓	11⅝	9.9	91.7	44.1	47.6	67.9	0.208
14	3	⅓	11⅝	13.49	107.9	46.0	61.9	76.9	0.218
15[a]	3	⅓	11⅝	17.58	97.0	47.0	50.0	72.0	0.351
16[a]	3	⅓	11⅝	39.55	156.9	57.5	99.4	107.2	0.398
17[b]	3	⅓	11⅝	17.58	132.4	49.0	83.4	90.7	0.211
18[b]	3	⅓	11⅝	39.55	226.0	63.4	162.6	144.7	0.243

[a] Tests No. 15 and 16 were made with 0.0003 in. crinkled aluminum foil.
[b] Tests No. 17 and 18 were made with 0.0003 in. plain aluminum foil.

varies with spacing and indicates a minimum value at about 0.37 in. The following example will illustrate how a point on this curve was obtained:

Assume a 3-in. space to be insulated. A spacing of 1/3 in. would require 8 sheets of aluminum. Number of spaces = 3 × 3 = 9. Height of space chosen, 11⅝ in. $C = 0.585$ Btu per hour per square foot per degree Fahrenheit (From Fig. 4). For practical purposes the conductance can be divided by the number of spaces, or

$$\frac{0.585}{9} = 0.065 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

The curve of Fig. 8 was found to have the same shape for different heights and overall space widths. Some experimental data on multiple spaces are given in Table 3.

The effect of height of the air space on the heat transfer coefficient appears to become constant above 35 in. Spaces below 4½ in. have not yet been investigated.

Conclusions

1. Plane bright metallic surfaces form an excellent heat insulation by reducing radiant heat transfer.
2. Insulation made up of such surfaces is usually very low in density, and is not seriously affected by moisture.
3. The optimum insulation spacing for a single air space is about 0.75 in.
4. The optimum insulation spacing for multiple air spaces is 0.37 in.
5. Increasing the height of a thin air space up to 35 in. increases the resistance to heat flow.
6. With conventional building materials about 75 per cent of the total heat transmission across enclosed vertical spaces is by radiation.

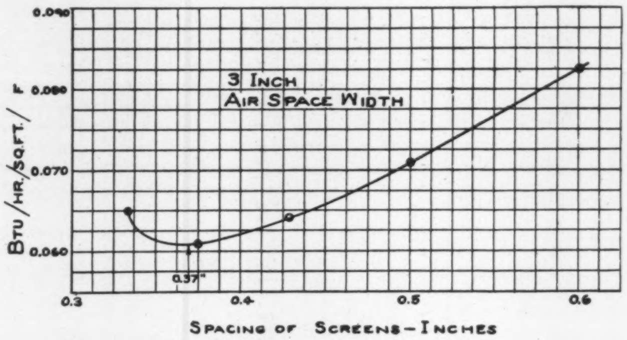

FIG. 8—OVERALL TRANSMISSION COEFFICIENTS FOR VARIOUS SPACINGS OF SCREENS

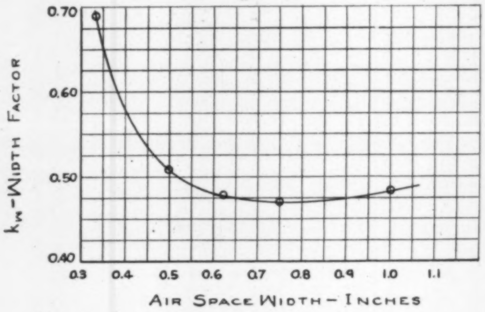

FIG. 9—CURVE SHOWING RELATION BETWEEN AIR SPACE AND WIDTH FACTOR

Appendix

An attempt was made to calculate the heat flow through specimens tested from a knowledge of the emissivity factors, height, spacing and temperatures. This was found to be very difficult by dividing the heat flow into its three component parts, conduction, convection and radiation. However, a purely empirical equation was set up based upon diffusion and radiation, as follows:

$$H_T = \frac{k_w \times k_h \Delta T^{1.266}}{(n+1)^{1.1}} + \frac{0.172 \times \frac{e_1\ e_2}{e_1 + e_2 - e_1\ e_2}\left[\left(\frac{T_1}{100}\right)^4 \quad \left(\frac{T_2}{100}\right)^4\right]}{(n+1)} \qquad (2)$$

where

H_T = total heat transmission in Btu per hour per square foot
k_w = air space width factor (Fig. 9)
k_h = air space height factor (Fig. 10)
ΔT = temperature difference hot to cold surfaces, degrees Fahrenheit
Exponent 1.266 taken from Surface Heat Transmission, by R. H. Heilman
Exponent 1.1 was found to suit the experimental data the best.
n = number of screens

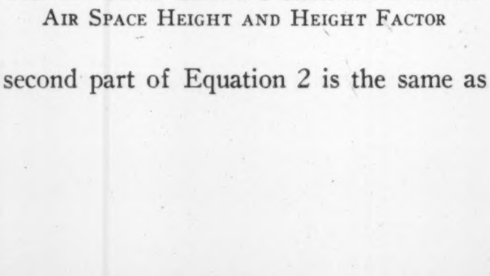

FIG. 10—CURVE SHOWING RELATION BETWEEN AIR SPACE HEIGHT AND HEIGHT FACTOR

The second part of Equation 2 is the same as Equation 1.

TABLE 4—TESTS MADE TO CHECK EQUATION 2

No. of Spaces	Width of Space (In.)	Height of Air Space (In.)	Measured Heat Input (Btu per hr per sq ft)	High Temp. (Deg Fahr)	Low Temp. (Deg Fahr)	ΔT (Deg Fahr)	MT (Deg Fahr)	Calculated Heat in put (Btu per hr per sq ft)	Variation between Calculated and Measured Heat Input (%)
1	½	9	17.58	83.0	47.2	35.8	65.1	17.23	−2
1	½	9	39.55	130.2	60.9	69.3	95.5	39.62	+0.2
2	½	9	17.58	116.8	49.0	67.8	82.9	18.09	+2.7
2	½	9	39.55	195.7	63.8	131.9	129.7	41.6	+5.3
3	⅓	11	17.58	127.9	49.3	78.6	88.6	16.98	−3.4
3	⅓	11	22.25	145.1	51.3	43.8	98.2	22.25	−3.1

The height and width factors were determined by taking the heat transferred by diffusion and temperature data for the one-inch space width from Fig. 7, and solving for k_h and k_w as a combined factor. By trial an arbitrary value for k_w of 0.48 was chosen and k_h was calculated for each height given on Fig. 7. Taking a height factor as a constant then k_w can be calculated for all widths. These two factors are plotted on Fig. 9 and Fig. 10.

This formula proved fairly satisfactory in calculating the total heat transfer in both single and multiple air spaces. An accuracy of 6 per cent was obtained by this method for the experimental range covered. Table 4 shows some tests made to check the formula.

Unit Air Conditioners

The unit air conditioner is a self-contained apparatus performing the same functions as the larger so-called central station equipment. It may be used for the air conditioning of printing plants, textile mills, candy manufacturing, packing and storage departments, cigar and cigarette making and seasoning departments, bakeries, process rooms necessary in the manufacture of pharmaceuticals and related products, and for any industrial plant or department in which production depends on, or quality is improved by, positive and accurate control over atmospheric conditions.

The unit air conditioner provides for humidification or dehumidification, air washing, heating or cooling, and uniform distribution of air, either with or without ducts, depending on the location of the units. Each unit is a complete air-conditioning plant in itself including spray nozzles, air re-heater, fan, pump, automatic control, etc. They are usually built in graduated standard sizes ranging in capacity up to 10,000 cfm each.

Advantages of Unit Air Conditioners

The principal advantages of such equipment are that the desirable results of the indirect humidification system can be applied to old buildings without installing duct work. Also where manufacturing processes are changed and departments are sub-divided or enlarged the units are easily moved or increased in number to suit the new requirements. Because of their portability they are useful for tenants of rented buildings.

Description and Operation

Unit air conditioners are manufactured in horizontal and vertical types and are thus adjustable to varying space requirements, being floor mounted or suspended. Water, steam and electrical connections are required to prepare them for operation. Electric power requirements generally range from 1 to 7 kw. Fan motors are from ⅛ to 5 hp. Water consumption varies from 40 to 200 gal per 12-hour period, according to the size of the unit.

An air-conditioning unit may be so designed that it will draw in air from the room or enclosure in which it is placed or from the outside, or by the aid of mixing dampers, may draw any predetermined mixture of inside and outside air. This entering air passes through one or more spray or mist chambers, in which space moisture is either added to or removed from the air in accordance with the temperature of the air and of the spray water. At the same time the air is washed substantially free of dust and solid matter in this chamber. The air next travels between the customary labyrinth eliminator plates to lose its entrained water. A suitably located fan assembly causes the air to flow over a heating unit before being delivered to the room. A motor-driven centrifugal pump or other means is used to deliver water from the tank built into the base of the unit to the spray heads, mist nozzles or other atomizing apparatus. In some makes of unit air conditioners, other means than spray nozzles are used to produce the required water spray or mist for the transfer of heat from air to water or vice versa. Automatic instruments control the condition of the leaving air by regulating the temperature of the spray water and also by varying the amount of steam supplied to the air heater. Such instruments usually are placed in the path of the return air as it enters the unit to be conditioned, as the entering air is a sample of room air. The spray water can be heated by an ejector or by a closed type heater.

Evaporative Cooling

In certain buildings in which a considerable amount of heat is generated by power driven machinery or from other sources involved in the manufacturing operations or during hot seasons when heat is transmitted into the building, evaporative cooling of the air, which occurs in the spray chamber of the unit, is sufficient to reduce the temperature of the air in the conditioned room from 5 to 20 deg. below the existing outdoor temperature. Though evaporative cooling is limited according to the relative humidity of the air entering the spray chamber of the unit cooling and dehumidifying type air conditioner, the general effectiveness contributes to the comfort of the workers and offers an efficient means of utilizing or disposing of the mechanical heat developed by machinery or of that given off from other sources.

Non-Evaporative Cooling

When it is desired to maintain conditions of temperature and humidity below those normally possible to obtain by evaporative cooling it is necessary to supply water to the unit at a temperature below the dew-point or saturation temperature of the entering air. The source of low temperature water may be a well, or the water may be cooled by refrigeration.

Output of Heating Units at Various Temperatures

By E. A. Allcut,[1] Toronto, Ontario
NON-MEMBER

An exposed heating unit located in the room or space to be heated is termed a "radiator." An enclosed heating unit is termed a "concealed radiator" or more properly a "convector." If, in the case of convectors, the flow of air over the heating unit takes place by natural convection, it is termed a "gravity convector," whereas if the air flow is produced by a fan or blower, it is termed a "fan convector."[2] *This paper deals with tests on heating units used as both radiators and gravity convectors as thus defined.—Ed.*

MOST of the work already described in connection with gravity convectors (concealed radiators) refers to steam as the heating medium. In Canada, however, the majority of homes and small buildings are heated with hot water. Some time ago the author made a series of tests on small heating units of various kinds, to determine by how much the heat output was reduced when using water at about 150 F, as compared with steam. The results were lower than had been expected and also varied somewhat with heating units of different form, so that the present series of tests was made for the purpose of obtaining some general information on this subject.

Heating units were selected of as many different forms as possible and were loaned by the manufacturers for this work. The enclosures used were of the dimensions recommended by them. Preliminary studies of the subject involved a consideration of the method of expressing the results. The output of a heating unit can be established fairly definitely by the usual procedure of testing with steam or hot water. On the other hand, the object of installing a heating unit is to provide heat for a room or building, and this involves not only the supply, but also the distribution of heat. Some manufacturers rate their heating units on the first basis and some on the second, and the situation is in urgent need of clarification as, for this reason, it is difficult to compare heating units of different design.

The author expresses no opinion as to which method is the better commercially, but came to the conclusion that the actual output in Btu per hour was the preferable basis for expressing the test results. The steam condensed, or quantity of water cooled, per hour indicates the actual amount of heat leaving the metallic surface, and previous experience had shown that this quantity was independent of changes in test conditions over comparatively wide ranges, so that similar results were obtained in rooms of different sizes, ceiling heights and window spaces. If this heat is not efficiently distributed, that is no fault of the heating unit itself, but of other conditions. Given a definite weight of air leaving the enclosure at a fixed temperature, the supply of heat to the room will be the same regardless of the type of heating unit inside the enclosure.

It is probable, also, that the floor dimensions of the room, the height of ceiling, the size and number of windows, direction, velocity and temperature of the air outside, all have a considerable influence on the distribution of heated air inside, thus introducing variables of unknown number and extent into the final results. For these reasons, the results are expressed deliberately in terms of heat passing over the heating surface, but readings of the velocities and temperatures at the stack outlet are also included.

Apparatus and Method of Testing

The tests were made in the Mechanical Engineering Laboratory of the University of Toronto, a shield of insulating material being placed at the back of the enclosure. There was also a ceiling at a height of 9 ft from the floor level to give an air flow approximating that obtained in practice. The layout is shown diagrammatically in Fig. 1.

Each heating unit was tested at several temperatures using both steam and hot water. A gas-fired boiler was used for both series of tests. This boiler could be reg-

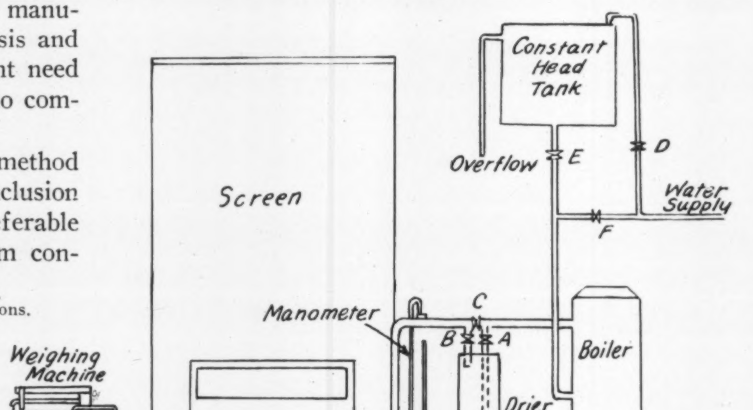

FIG. 1 — DIAGRAM OF APPARATUS

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[1] Professor of Mechanical Engineering, University of Toronto.
[2] Proposed definitions of A. S. H. V. E. Committee on Definitions.

ulated to supply the amount of heat necessary to maintain the required temperature difference between the heating unit and the air entering the enclosure.

For the steam tests, the water was kept at about the center of the boiler gage glass and the steam was admitted to the separator by opening valves *A* and *B* and closing valve *C*. The water was drained from the separator at intervals during each test, so that dry steam alone was admitted to the heating unit. All steam and hot water pipes were covered to avoid radiation and condensation losses. This method was preferred to that of superheating the steam because of its greater simplicity and closer approximation to the conditions actually existing in gravity convector installations. Its accuracy has been proved by previous experience and by comparing the tests made in this way with results on the same or similar heating units tested by the superheated steam method in other laboratories. There was no appreciable difference in the results obtained.

The condensate was drained from the heating unit at frequent intervals during each test, so that the heating surfaces were kept dry. It was then collected in a weighing tank, observations being taken every ten or fifteen minutes. In each case, no test was started until successive readings showed that a constant amount of steam was being condensed, so that conditions were steady. Each test was then run for about 90 min, as it had been found by previous experience that this period was ample for the small heating units tested. The inlet and outlet temperatures were taken both by thermometers and thermocouples, and the agreement generally was satisfactory. The amount of disagreement is shown in the tables of test results. The steam pressure was indicated by a mercury manometer.

Considerable difficulty was experienced in making hot water tests, as it was by no means easy to maintain a constant flow of water through the heating units. Finally, the layout shown in Fig. 1 was adopted and was used throughout the tests. The same boiler was used as before, but it was now filled with water. The water was supplied from the city main to a constant head tank, by opening valves *D* and *E* and closing valve *F*. The water in the tank was kept up to the level of the overflow and the quantity supplied to the system was regulated by opening and closing valve *H*. This varied the temperature drop across the heating unit, and at the same time preserved a full head of water on the system. The mean temperature was regulated by increasing or reducing the quantity of gas supplied to the burners. The water thus heated left the boiler through valve *C*, (which was now opened, valves *A* and *B* being closed) passed through the heating unit and was measured by weighing. Thermocouples again were used in addition to the thermometers, and at first the bare wires were placed in the water stream, but erratic results were obtained. It was found, after investigation, that these were due to stray currents. The thermocouples were therefore placed in glass tubes filled with oil and it was found that with this method the inconsistencies disappeared.

The heat outputs for the steam tests were then plotted against the temperature differences between the steam and the air entering the heating unit, and a curve was drawn through them of the form

$$\left(\frac{215-65}{\text{steam temperature}-\text{air temperature}}\right)^{1.3}$$

The hot water tests were plotted in a similar manner, the mean temperature of the heating unit being considered to be the arithmetic mean between the inlet and outlet water temperatures. It is questionable whether this should be adopted in all instances as in most cases of heat transmission the logarithmic mean is taken as the correct temperature, but the foregoing method was employed in accordance with generally accepted practice. The difference between the arithmetic and logarithmic mean temperatures for the test conditions was generally less than one degree Fahrenheit.

In most heating units the hot water tests fell on a curve below that drawn for steam, and in such cases a similar curve was drawn through the points obtained by the hot water tests. The ratio of the heat output with hot water to that with steam at the standard temperature difference of 105 deg is given for each heating unit. This indicates the extent to which the output with hot water falls short of that with steam with the same atmospheric conditions.

Form of Enclosure

The enlargement of the Royal Ontario Museum provided somewhat unusual conditions for enclosed heating units, the room being very large, the ceilings high and the windows nearly 12 ft wide. Stack heights of 8 ft were required to enable the air to be discharged above the specimen cases.

Experiments were made with the different types of heating units placed in an adjustable enclosure, so that heat outputs could be measured under different conditions, as follows:

1. The height of the air inlet was reduced from 13 in. to 6 in., the equivalent heating surface being raised from 61.1 to 68.2 sq ft, an increase of 11.6 per cent. Thus, it appears to be inadvisable to increase the air inlet opening beyond a certain limit.
2. The original plans called for the air outlet opening in some cases to be arranged as in *AH2* (Fig. 2), the object being to cover the entire width of the window with a screen of warm air. On the other hand, this increase of area decreases the outlet velocity of the air and therefore reduces its effectiveness.

   The results obtained with the top of the enclosure set at different angles are shown in Fig. 2. The air velocity curves show that there is practically no upward movement of air outside the vertical stack opening, and that on the left side there is actually a down draft. The equivalent heating surfaces for the three tests were:

   AH2 (10 ft 11 in. wide) 68.2 sq ft (45 deg)
   AH3 (8 ft 6 in. wide) 68.7 sq ft (22.5 deg)
   AH4 (6 ft 8½ in. wide) 71.4 sq ft (Parallel)

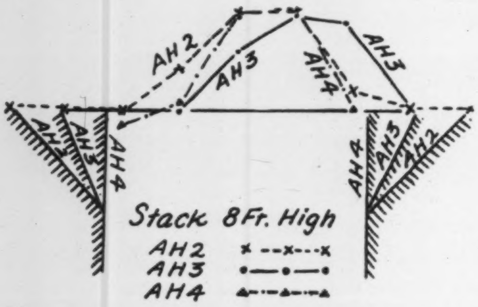
   Thus, the parallel stack is to be preferred.

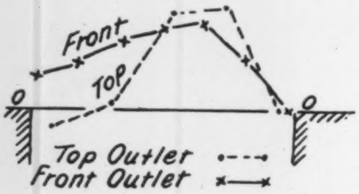
   These tests were all made on days when there was a southwest wind of 3 to 4 mph, so that they are comparative.
3. The uneven distribution of air in Fig. 2 attracted attention to the direction of air flow from the enclosure. It is evident that when the heating unit is placed under a window, the downward current of cold air will tend to oppose the upward current of warm air from the enclosure. By placing a baffle over the top and directing the flow of air horizontally, this tendency is reduced and the figures in Table 1 indicate that the improvement obtained from this change was about 10 per cent.

It was noticed with the vertical outlet, that the air

Table 1—Increased Output Due to Baffle Placed Over Top of Enclosure

Test No.	Outlet	Equivalent Heating Surface (Square Feet)	Increased Output (Per Cent)
AH4	Top	71.4	....
AH6	Front	78.4	9.8
AK1	Top	84.2	....
AK2	Front	92.1	9.4
AL1	Top	84.6	....
AL2	Front	93.5	10.5





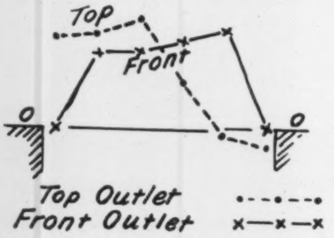


Fig. 2 (Top)—Results Obtained with Top of Enclosure Set at Different Angles (*AH2*, *AH3*, *AH4*)

Fig. 3 (Middle)—Distribution of Air Across Top and Front Outlets (Down Draft on Left)

Fig. 4 (Bottom)—Distribution of Air Across Top and Front Outlets (Down Draft on Right)

pulsated somewhat, so that correct readings of its velocity were difficult to obtain. These pulsations were almost entirely stopped when the horizontal outlet was substituted, the air being emitted in a steady stream. The distribution of air across the outlet with top and front outlets respectively, is shown in Figs. 3 and 4. These indicate a more uniform distribution for the front outlet, in addition to the steadier flow. The top outlet has a down draft on the left of Fig. 3 and the right of Fig. 4.

Steam and Hot Water Tests

These tests were made on small convectors of approximately 20 and 40 sq ft (equivalent) rating, respectively, and of as many different forms as it was possible to obtain. There is no intention of comparing the relative merits of the different convectors, save insofar as their design and general dimensions affect the ratio of the output on hot water to that on steam.

The direct or prime heating surface in each case is considered to be that area which is in contact with steam or water on one side and air on the other. The remainder of the measured area is indirect or secondary heating surface. Wherever practicable, the temperature of the direct heating surface and that of the indirect surface farthest from the heating medium were measured.

Heating Units BC and BD

These were both of the ordinary cast-iron type and were tested with steam and hot water, respectively, both bare and enclosed. Heating unit *BC* was tall and thin, and *BD* was low and wide. See Table 2. Both were unpainted and the results obtained are indicated by Figs. 5 and 6.

Table 2—Physical Characteristics of Heating Units *BC* and *BD*

	BC	*BD*
Columns	3	6
Sections	6	6
Height (inches)	38	20
Length (inches)	15	15
Direct heating surface (square feet)	18.7	17.6
Indirect heating surface (square feet)	0	0
Air inlet area (square feet)	0.437	0.437
Air outlet area (square feet)	0.75	0.742

The curves show that, whether bare or enclosed, the output with hot water, in both cases, follows the same curve as that for steam. The relationships given in Table 3 are taken from Figs. 5 and 6.

Table 3—Results Obtained With Heating Units *BC* and *BD*

Heating Unit	*BC*		*BD*	
State	*Bare*	*Enclosed*	*Bare*	*Enclosed*
Btu per hour at				
a. 150-deg temp diff....	5,200	4,000	4,920	4,120
b. 105-deg temp diff....	3,300	2,520	3,100	2,600
c. 55-deg temp diff....	1,450	1,080	1,350	1,110
$\frac{b}{a}$ per cent............	63.5	63.0	63.0	63.1
$\frac{c}{a}$ per cent............	27.9	27.0	27.2	26.9

The air velocities (feet per minute) were obtained at the left, center and right ends of the stack outlet, respectively, and indicate that the highest velocities occur at the right, or entrance, side of the heating unit (Fig. 1).

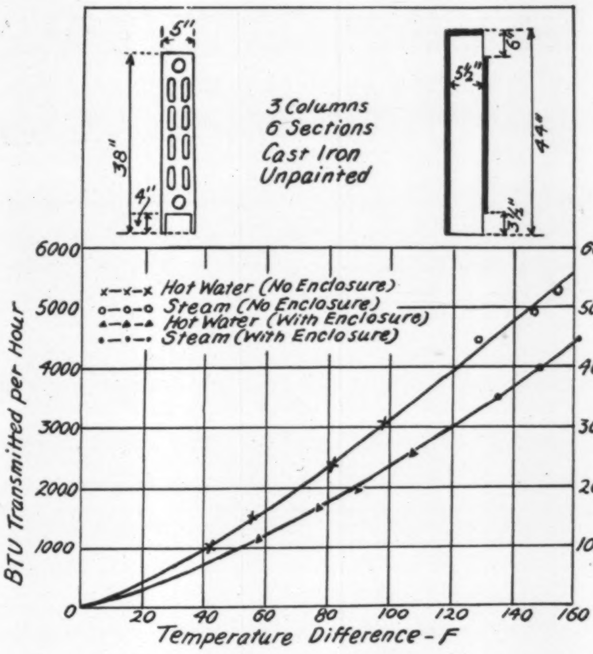


FIG. 5—RESULTS OBTAINED WITH HEATING UNIT *BC*

Heating Units AO and AX

These consisted of aluminum plates or fins, into which tubes were expanded, the whole assembly being painted black. The details are given in Table 4.

TABLE 4—DETAILS OF HEATING UNITS *AO* AND *AX*

	AX	*AO*
Number of tubes	2	6
Diameter of tubes (inches)	1½	1½
Length of tubes (inches)	19½	19½
Number of fins	45	88
Area of each fin (inches)	6¾ x 4	10¾ x 4
*Direct heating surface (square feet)	1.2	3.6
*Indirect heating surface (square feet)	14.6	45.7
Air inlet area (square feet)	0.482	0.482
Air outlet area (square feet)	0.878	0.878
Height of feet (inches)	4½	4½
Thickness of fins (inches)	0.041	0.041

*These are based on tube and fin surface only. If the entire surface of the headers is included, the direct heating surface of *AX* is increased by 0.9 sq ft and that of *AO* by 2.9 sq ft. The coefficients given in Fig. 13 must be reduced correspondingly.

TABLE 5—RESULTS OBTAINED WITH HEATING UNITS *AX* AND *AO*

	AX	*AO*
Btu per hour at		
a. 150-deg temp diff	3,920	7,400
b. 105-deg temp diff	1,950	4,250
c. 55-deg temp diff	800	1,850
$\frac{b}{a}$ per cent	49.8	57.5
$\frac{c}{a}$ per cent	20.4	25.0
Per cent of steam curves at		
d. 105-deg temp diff	79.6	91.4
e. 55-deg temp diff	76.2	92.5

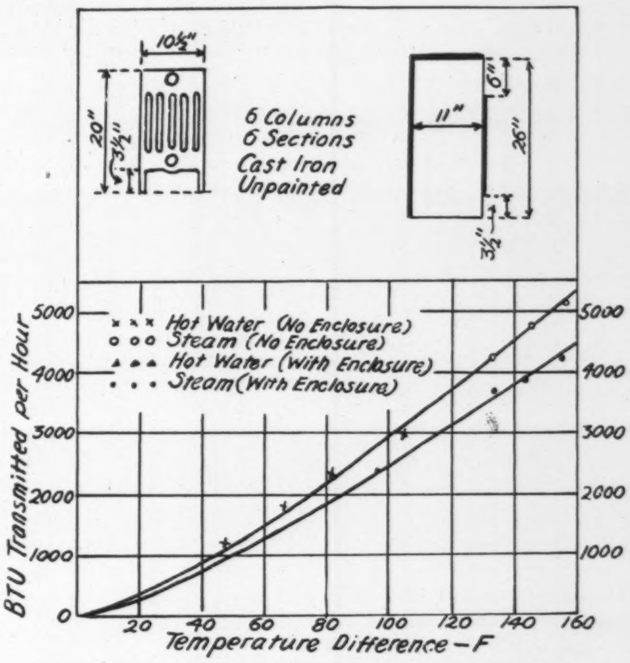


FIG. 6—RESULTS OBTAINED WITH HEATING UNIT *BD*

The same enclosure was used for both heating units. The results obtained are illustrated by Fig. 7. The relationships given in Table 5 were taken from the curves of Fig. 7. These figures show that the hot water output at a temperature difference of 105 deg is from 10 to 20 per cent less than that anticipated from the steam tests, and at the lower temperature the output may be as much as 24 per cent lower.

The air velocities were taken at the left, center and right of the stack outlet, positions 1, 3 and 5 being at the top of the grille, and 2, 4 and 6 at the bottom. The zero figures given for the lower positions indicate in all cases that the velocities were too low to work the

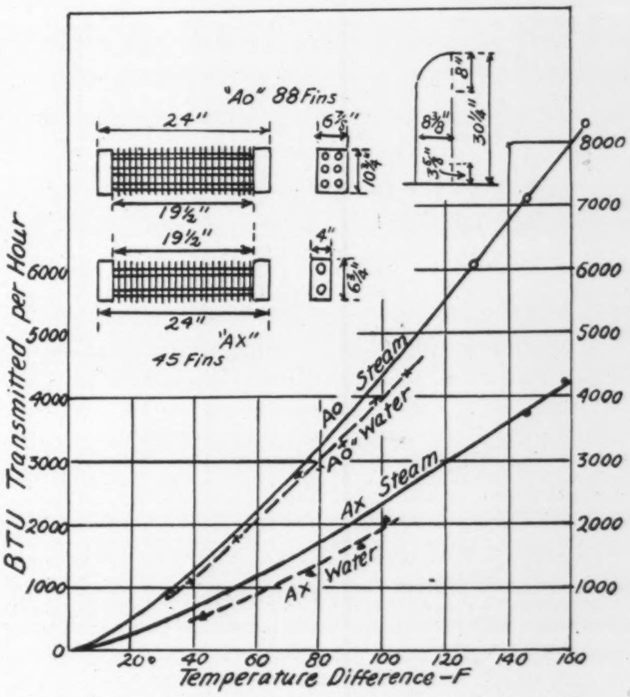


FIG. 7—RESULTS OBTAINED WITH HEATING UNITS *AO* AND *AX*

anemometer. It is evident that most of the air comes out at the upper edge, so that the grille depth might be reduced with advantage.

Heating Units AM and AN

These were both composed of copper tubes expanded into aluminum fins. The latter were separated by means of spacers, which formed an integral part of each fin. The fins were smaller and were more closely spaced than in the previous cases and were shrouded at the edges by thin metal strips. The finish was left bright. The dimensions are given in Table 6. The test results are indicated by Fig. 8 and the figures in Table 7 were obtained from these curves.

TABLE 6—DIMENSIONS OF HEATING UNITS *AM* AND *AN*

	AM	*AN*
Number of tubes	2	3
Diameter of spacers (inches)	0.53	.53
Length of tubes (inches)	26½	32¼
Number of fins	155	195
Area of each fin (inches)	2 x 3³⁄₁₆	2 x 5³⁄₁₆
Direct heating surface (square feet)	0.95	1.40
Indirect heating surface	12.8	26.5
Air inlet area (square feet)	0.71	0.86
Air outlet area (square feet)	0.61	0.73
Thickness of fins (inches)	0.016	0.016

TABLE 7—DATA FOR HEATING UNITS *AM* AND *AN* TAKEN FROM FIG. 8

	AM	*AN*
Btu per hour at		
a. 150-deg temp diff.	4,320	7,520
b. 105-deg temp diff.	1,950	3,920
c. 55-deg temp diff.	720	1,350
$\frac{b}{a}$ per cent	45.2	52.1
$\frac{c}{a}$ per cent	16.7	18.0
Per cent of steam curve at		
d. 105-deg temp diff.	71.0	82.5
e. 55-deg temp diff.	61.0	67.5

The loss of output in this case is greater than before, being 18 to 29 per cent at 105 deg and 33 to 39 per cent at 55 deg. The outlet grille was too narrow for double anemometer readings so that air velocities for the left, center and right positions were recorded. The highest velocities were obtained in the center. It should be noted that the outlet area of the enclosure was less than the inlet area.

Heating Unit AY

This was similar in form to heating units *AM* and *AN*, but was made entirely of cast iron, unpainted, the water or steam space being a rectangular cored hole passing through the center of the casting, with fins radiating on each side. The dimensions are given in Table 8. The test results are indicated by Fig. 9 and figures obtained from the curves are given in Table 9.

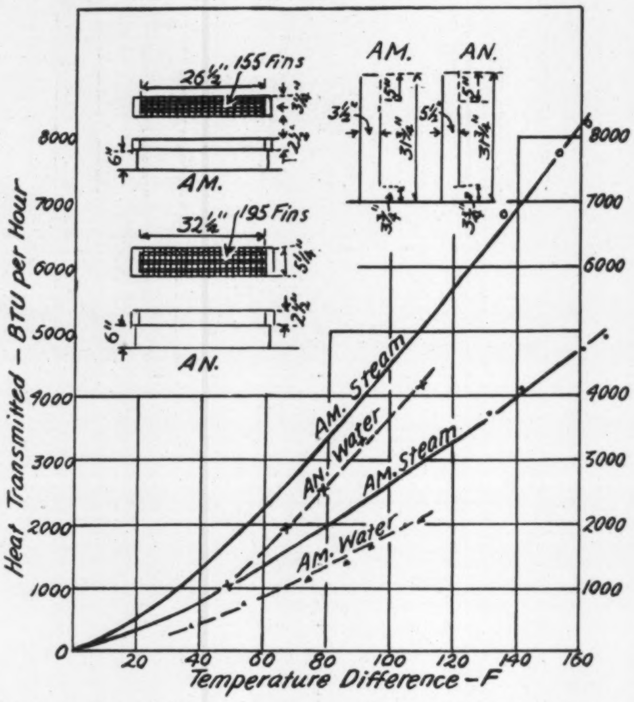


FIG. 8—RESULTS OBTAINED WITH HEATING UNITS *AM* AND *AN*

The losses in this case are 23 and 27 per cent at temperature differences of 105-deg and 55-deg, respectively. Other tests made on similar heating units with larger and smaller numbers of fins gave a similar relationship. The air velocities, as before, were at the left, center and right of the outlet, the highest velocities being obtained at the inlet end.

It was observed generally that the higher heating units gave outputs on hot water more closely approximating to those on steam than did the lower ones. To check this, two heating units similar to *AY* were arranged in parallel, one above the other, so that the steam or water entered at the right end of both heating units and came

TABLE 8—DIMENSIONS OF HEATING UNIT *AY*

Length of radiator (inches)	28
Number of fins	100
Size of fins (inches)	3¾ x 3½
Direct heating surface (square feet)	1.33
Indirect heating surface (square feet)	17.5
Air inlet area (square feet)	0.630
Air outlet area (square feet)	0.845
Thickness of fins (inches)	0.063

TABLE 9—DATA FOR HEATING UNIT *AY*

Btu per hour at		
a. 150-deg temp diff.	4520	
b. 105-deg temp diff.	2200	
c. 55-deg temp diff.	890	
$\frac{b}{a}$ per cent	48.7	
$\frac{c}{a}$ per cent	19.7	
Per cent of steam curve at		
d. 105-deg temp diff.	77.2	
e. 55-deg temp diff.	73.0	

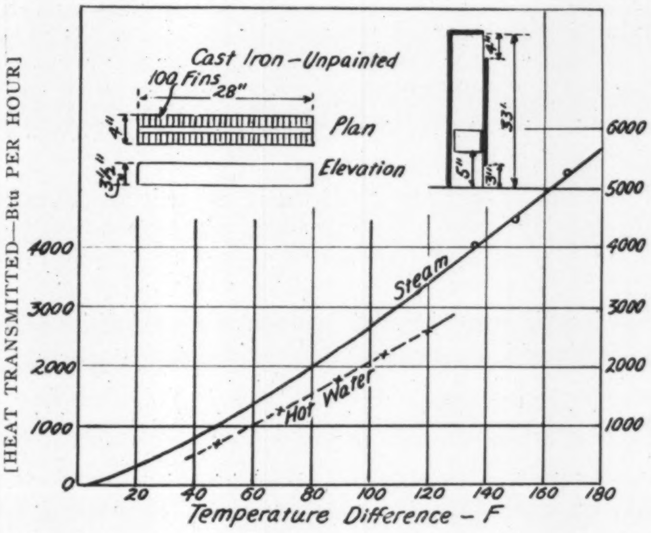


FIG. 9—RESULTS OBTAINED WITH HEATING UNIT *AY*

out on the left side. This increased the total output to 14.5 per cent more than that of the single heating unit *AY* and made no difference in the ratio of hot water to steam output, which remained at about 77 per cent at a temperature difference of 105 deg.

The single *AY* was then placed above and in series with the other two so that the steam or water entered at the right end of *AY*, came out on the left and then traversed the other two in parallel, from left to right. The output was then 18.4 per cent above that of the single heating unit *AY*, but the ratio of hot water to steam output was raised to 95 per cent, a considerable improvement.

The next modification was to connect the heating units in such a way that there was a vertical bank of three heating units in the enclosure, all operating in series. This arrangement increased the steam output at 150-deg temperature difference by 4.5 per cent as compared with the previous case. The transmission with hot water, however, was not increased, so that the output ratio was reduced to 90.2 per cent.

The final tests were made with two heating units similar to *AY* arranged in series and placed one above the other. The heat transmission on steam was slightly greater than that with the two heating units in parallel, but that on hot water was improved considerably. At a temperature difference of 105 deg the output on hot water was increased by 18 per cent, and the ratio of the heat transmitted with hot water to that with steam was raised from 77 per cent to 93.2 per cent.

Heating Units AR, AS and BB

These were all of similar construction, being composed of bent copper tubes with pressed sheets clamped and welded to them to form chimneys, as shown in Figs. 10 and 11. Heating units *AR* and *BB* were of similar ratings, but different heights, and this comparison was made to indicate the influence of overall dimensions, particularly height, on the relationship between outputs on steam and water. The finish in all cases was black and the dimensions are given in Table 10. The results obtained are given in Figs. 10 and 11. The data in Table 11 have been taken from the curves.

It will be noted that heating unit *AR* is the only indirect unit in which the hot water output follows the same curve as that for steam, but that the other two of the same type lose comparatively little. It is also noteworthy that these heating units maintain this relationship as the temperature difference is reduced, whereas the other types lose more under those conditions.

The air velocity figures show that the velocity reaches a maximum in the center of the air outlet. As in previous cases, the velocities at 2, 4 and 6 near the bottom of the air outlet are very small. A smaller enclosure could probably have been used with advantage in the case of heating unit *AS*.

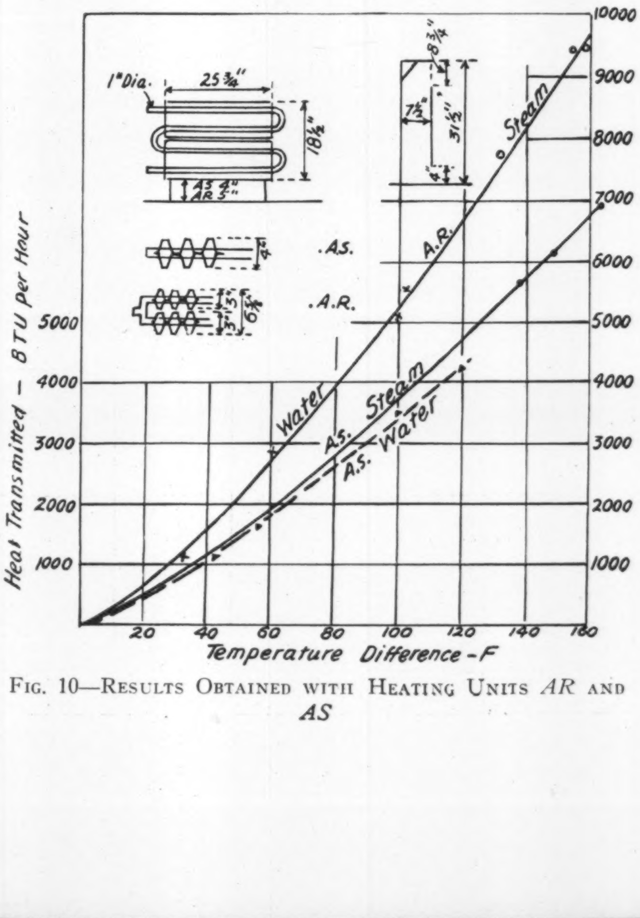


FIG. 10—RESULTS OBTAINED WITH HEATING UNITS *AR* AND *AS*

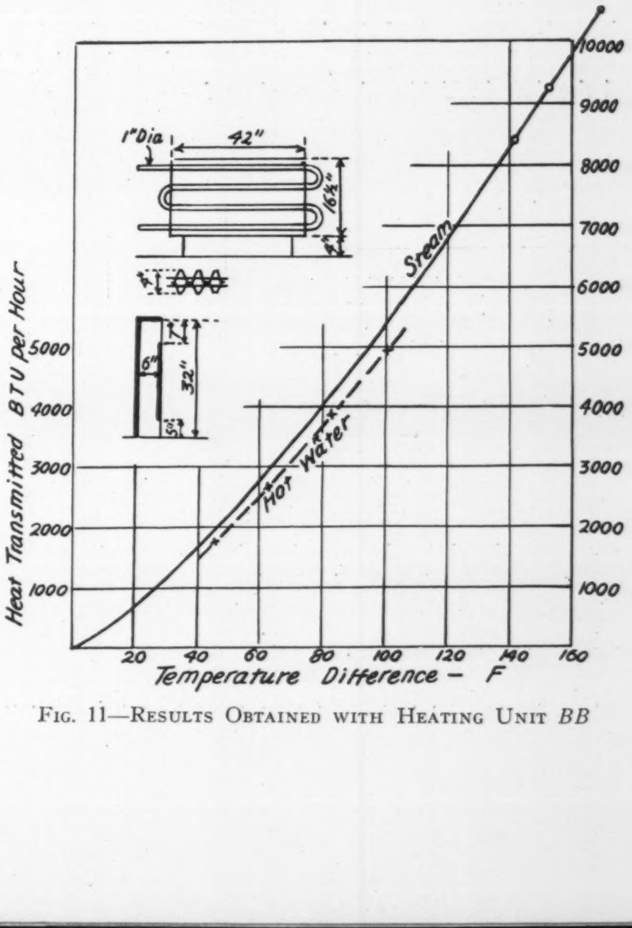


FIG. 11—RESULTS OBTAINED WITH HEATING UNIT *BB*

TABLE 10—DIMENSIONS OF HEATING UNITS *AS, AR* AND *BB*

	AS	*AR*	*BB*
Diameter of tubes (inches)	1.0	1.0	1.0
Height of chimneys (inches)	18½	18½	16½
Number of sections	18	18	30
Rows of tubes	1	2	1
Direct heating surface (square feet)	2.83	5.7	4.15
Indirect heating surface (square feet)	41.13	64.8	59.34
Air inlet area (square feet)	1.10	1.10	1.74
Air outlet area (square feet)	1.69	1.69	2.43

TABLE 11—DATA FOR HEATING UNITS *AS, AR* AND *BB*

	AS	*AR*	*BB*
Btu per hour at			
a. 150-deg temp diff	6120	8900	9150
b. 105-deg temp diff	3600	5600	5250
c. 55-deg temp diff	1600	2410	2300
$\frac{b}{a}$ per cent	58.9	63.0	57.5
$\frac{c}{a}$ per cent	26.2	27.0	25.2
Per cent of steam curve at			
d. 105-deg temp diff	93.5	100	91.3
e. 55-deg temp diff	95.6	100	93.2

Heating Units BE and BF

These heating units consist of one and two rows, respectively, with 5 finned tubes in each, arranged diagonally in the enclosure, as shown in Fig. 12. The peculiar form of the end castings is designed to permit easy removal of the heating unit from the enclosure.

The fins consist of spirals of copper or brass surrounding the tubes, the whole of the surface being coated with lead or solder, making good metallic contact at the joints. The dimensions are given in Table 12.

The dampers supplied with the enclosures were kept fully open in all tests. The results are given in Fig. 12 and the data in Table 13 were taken from the curves.

TABLE 12—DIMENSIONS OF HEATING UNITS *BE* AND *BF*

HEATING UNIT	*BE*	*BF*
Number of tubes	5	10
Diameter of tubes (inches)	¾	¾
Length of finned surface (inches)	29½	29½
Diameter over fins (inches)	1½	1½
Direct heating surface (square feet)	3.5	6.2
Indirect heating surface (square feet)	16.5	32.9
Air inlet area (square feet)	1.21	1.21
Air outlet area (square feet)	1.10	1.10

Conclusions

An analysis of the figures given in the preceding paragraphs gives no definite indication of the features of design that are necessary for obtaining the best results on hot water. The output ratio appears to be independent of the ratio of direct to indirect heating surface.

It is noticeable, however, that heating units such as *AO, AS, AR,* and *BB,* in which the air passes more than once over a direct heating surface, give the highest output ratios. Also, the heating units with dull black finish appear to be better than those left in the bright condition, but the number and variety of experiments is not sufficient to prove this.

The velocities of water and steam, respectively, through the heating units were calculated. The former vary from 1.03 to 17.97 fpm at a temperature difference of 105 deg, and the latter from 54.6 to 955 fpm. Again no relationship was observed between these velocities and the ratio of heat outputs.

The coefficients of heat transmission were then computed for each heating unit, the figures for steam being plotted in Fig. 13 from 110 F upwards, and for water from 110 F downwards. This division is purely arbitrary but corresponds fairly well to the usual working range in each case. The variation in these coefficients is remarkable, ranging as it does from over 2.0 to less than

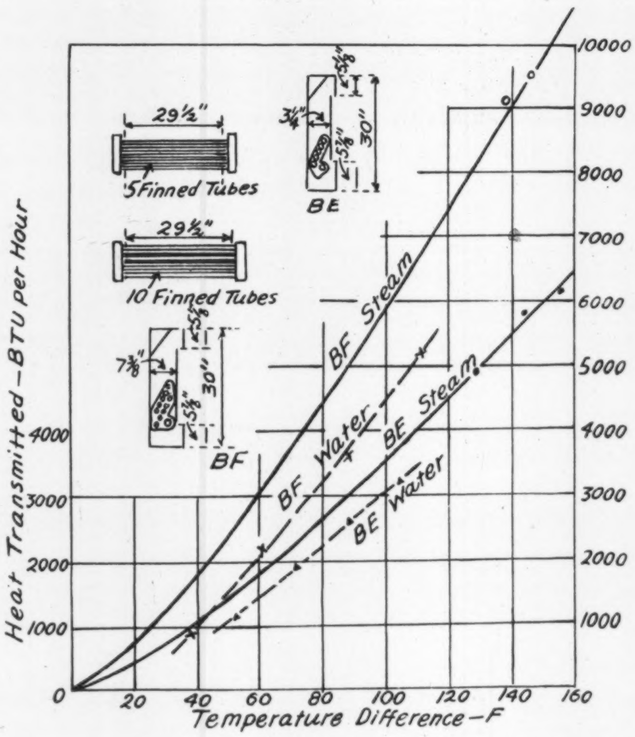


FIG. 12—RESULTS OBTAINED WITH HEATING UNITS *BE* AND *BF*

TABLE 13—DATA FOR HEATING UNITS *BE* AND *BF*

HEATING UNIT	*BE*	*BF*
Btu output at		
a. 150-deg temp diff	6000	9880
b. 105-deg temp diff	3250	4800
c. 55-deg temp diff	1260	1820
$\frac{b}{a}$ per cent	54.2	48.6
$\frac{c}{a}$ per cent	21.0	18.2
Per cent of steam curve at		
d. 105-deg temp diff	86.2	77.5
e. 55-deg temp diff	77.8	68.5

0.9 Btu at the standard temperature difference of 150 deg. It is also evident that the higher coefficients are associated with the greater differences between steam and hot water outputs and that heating units *AO* to *AR* (Fig. 13) having the higher output ratios, have very low heat transmissions.

The ordinary cast-iron radiators *BC* and *BD*, when tested bare, have almost exactly the same heat transmissions per square foot for one degree temperature difference, in spite of their different proportions, and are included for comparative purposes.

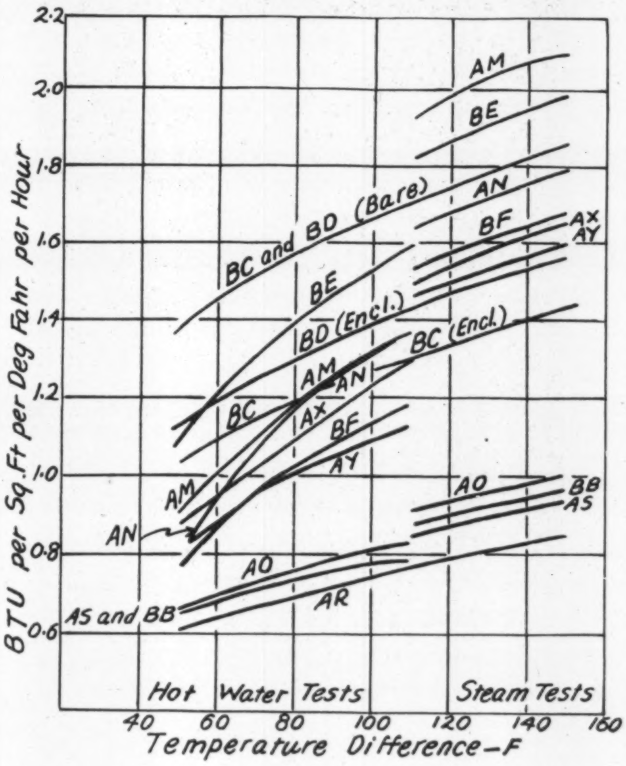


FIG. 13—COEFFICIENTS OF TRANSMISSION OF HEATING UNITS AT VARIOUS TEMPERATURE DIFFERENCES

The results may, therefore, be summarized as follows:

1. The output of a gravity convector (concealed radiator) varies with the area of the air inlet which may be too large to give the best efficiency.

2. The best form of stack is that with vertical sides. Increasing the area at the top may reduce the output considerably.

3. When placed under a window, down draft and consequent inefficiency may be avoided by putting the air outlet in the front of the enclosure instead of at the top.

4. The heat output with hot water is usually less than that expected from steam tests. The amount of loss appears to depend more on the disposition of the direct heating surface than on the relative amounts of direct and indirect surface. It may also bear some relation to the finish of the surface, whether bright or dull.

5. The distribution of air across the outlet is seldom uniform. In most cases the greatest velocity is obtained on that side of the heating unit at which the steam or water enters. Most of the air is emitted from the top three or four inches of the outlet slot or grille.

6. The difference between the temperatures of the direct and indirect heating surfaces were generally less than 1 F, except in the case of *AY*, where consistent differences of 1½ to 2 deg were observed. It follows that if the direct heating surface is the more efficient, this is probably due to the fact that in most cases the air flow is perpendicular to the direct heating surface and parallel to the indirect heating surface.

7. The coefficient of heat transmission varies considerably in heating units used as gravity convectors, but generally the higher ratios of hot water to steam outputs are obtained with the surfaces that are lightly loaded from a heat transmission standpoint.

This research was conducted under the auspices of the School of Engineering Research of the University of Toronto. The author's thanks are due to the authorities of the University and to Prof. R. W. Angus for the facilities provided; also to E. B. Hymmen, who was largely responsible for the actual experimental work, and to D. T. Hewson, who assisted in this work.

Output of Wall or Cabinet Convectors

It has been the practice to state the output of direct radiators in terms of the *equivalent square foot* of heating surface. With the development of convection-type heating units (extended surface units) the same procedure has been followed. However, an extended surface heating unit is entirely different structurally and physically from that of a direct radiator and since it has no physical measurement corresponding to the square foot of heating surface of a direct radiator, many engineers believe that heating units of this type should be rated in Btu output, rather than in equivalent square feet of heating surface.

Convection-type heating units usually maintain room temperatures equivalent to those produced by exposed heating units with less steam consumption. This characteristic of the performance of convection-type heating units has been termed *heating* effect. Heating effect is attributed to (1) the minimization of radiant emission (2), the fact that the air which passes upward through a convection-type heating unit is drawn by gravity from the coolest strata in the room, *i.e.*, that at the floor level and (3) the heated air emitted from the air outlet of the enclosure is generally projected out into the room instead of rising directly to the ceiling.

The general effect of an enclosure placed about a heating unit of the type used as a direct radiator is to decrease the amount of steam that the unit will condense per hour and thus to reduce the amount of heat given off by the unit. The results of investigations at the University of Illinois[1] indicate that only where the enclosure is made much higher than the heating unit, thereby reducing a chimney effect, is the condensation rate equal to or greater than that of an identical radiator unenclosed. The effects produced by an enclosure are dependent upon its design, so that the following results may be produced:

1. Better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.

2. The lessened steam consumption may not materially change the heating performance.

3. The enclosed radiator (convector) may inadequately heat the space.

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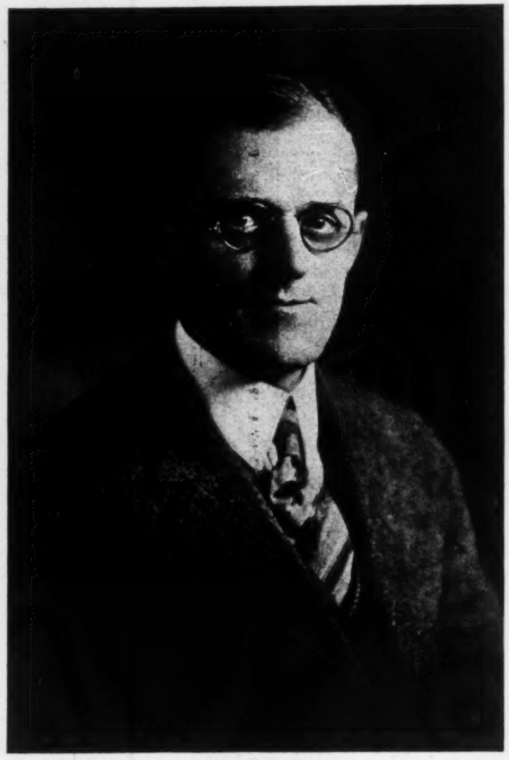
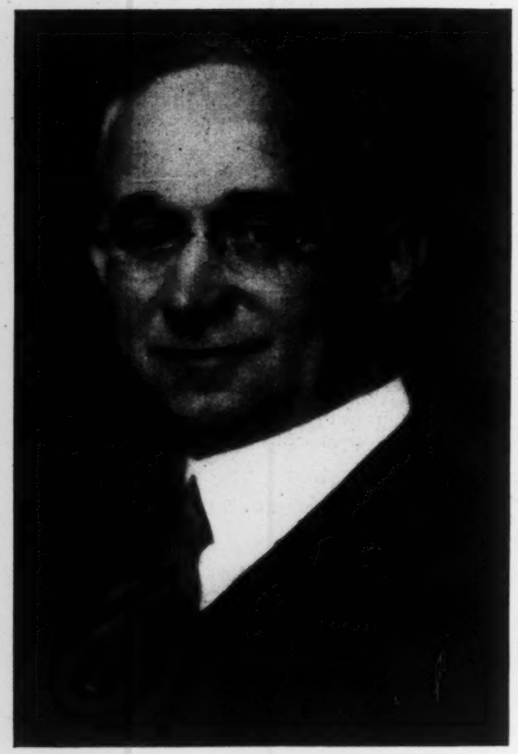
[1] Effect of Enclosures on Radiator Performance, by A. P. Kratz and M. K. Fahnestock (A. S. H. V. E. TRANSACTIONS, Vol. 33, 1927).

Candidates for 1932

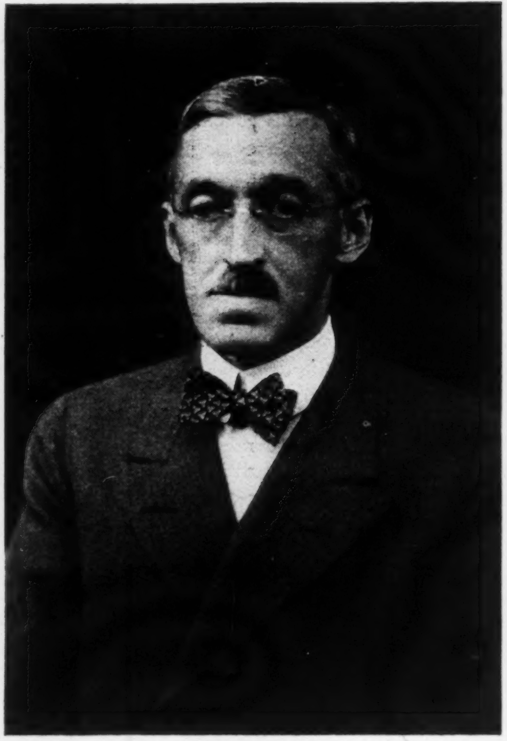
Prof. F. B. Rowley
Minneapolis



W. T. Jones
Boston



C. V. Haynes
Philadelphia



F. D. Mensing
Philadelphia

Candidates for Council

PROF. F. E. GIESECKE
Austin, Texas

PROF. G. L. LARSON
Madison, Wis.

J. F. McINTIRE
Detroit

NOMINATIONS FOR 1932

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1932, takes pleasure in submitting the following list of nominees:

For President:

F. B. Rowley, Minneapolis, Minn.

For First Vice-President:

W. T. Jones, Boston, Mass.

For Second Vice-President:

C. V. Haynes, New York, N. Y.

For Treasurer:

F. D. Mensing, Philadelphia, Pa.

For Members of the Council:

Three-Year Term

F. E. Giesecke, College Station, Texas.
G. L. Larson, Madison, Wis.
J. F. McIntire, Detroit, Mich.
W. E. Stark, Cleveland, Ohio

Respectfully submitted,

NOMINATING COMMITTEE

R. C. Bolsinger, *Chairman*

Chapters	*Representative*
Cleveland	H. M. Nobis
Illinois	H. M. Hart
Kansas City	N. W. Downes
Massachusetts	David Moulton
Michigan	J. F. McIntire
Minnesota	M. S. Wunderlich
New York	Russell Donnelly
Western New York	F. H. Burke
Ontario	H. J. Church
Pacific Northwest	E. O. Eastwood
Philadelphia	R. C. Bolsinger
Pittsburgh	F. C. McIntosh
St. Louis	C. A. Pickett
Southern California	R. L. Gifford
Wisconsin	E. A. Jones

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. 9—Section 2. The Nominating Committee shall consist of a member designated by each Chapter, which shall effect its own organization and elect its own Chairman.

Section 3. The Committee may meet at the call of the Chairman, for discussion, but the official meeting shall be held at the Semi-Annual Meeting of the Society, at which time the candidates for each of the offices of the Society to be filled at the next Annual Meeting shall be selected by a majority vote of the members of the Committee present and voting. Five shall constitute a quorum.

Section 4. The Committee shall first secure the consent of all candidates selected, and shall, if possible, announce the names of the candidates at the Semi-Annual Meeting, but in any case the names of the candidates shall be certified to the Secretary of the Society at least four months before the next Annual Meeting. The Secretary shall publish these names in the October issue of The Journal.

Art. 10.—*Section* 3. The Secretary shall prepare ballots with the names of all candidates and forward them to the members at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election.* The election is governed by the By-Laws for the election of officers, with the single exception, that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the government of the Research Laboratory, adopted at the January, 1919, Meeting, and revised January, 1930, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1932:

D. E. French, Philadelphia, Pa.
F. E. Giesecke, College Station, Texas
A. P. Kratz, Urbana, Ill.
G. L. Larson, Madison, Wis.
L. A. Harding, Buffalo, N. Y.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

Section I—Organization

Research Committee

1. There shall be a standing committee known as the Research Committee, consisting of fifteen members, each serving for three years, and five retiring each year.

a. The Council shall nominate previous to October first of each year five members to fill the vacancies of those retiring at the next Annual Meeting.

b. The nominations made by the Council shall be published in the October issue of the Society's Journal.

c. Prior to December first of any year, any ten members, over their own signatures, may nominate one or more additional members of the Research Committee, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

d. The election shall otherwise conform to the regulations provided for the election of officers of the Society.

e. Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

Candidates for Committee on Research

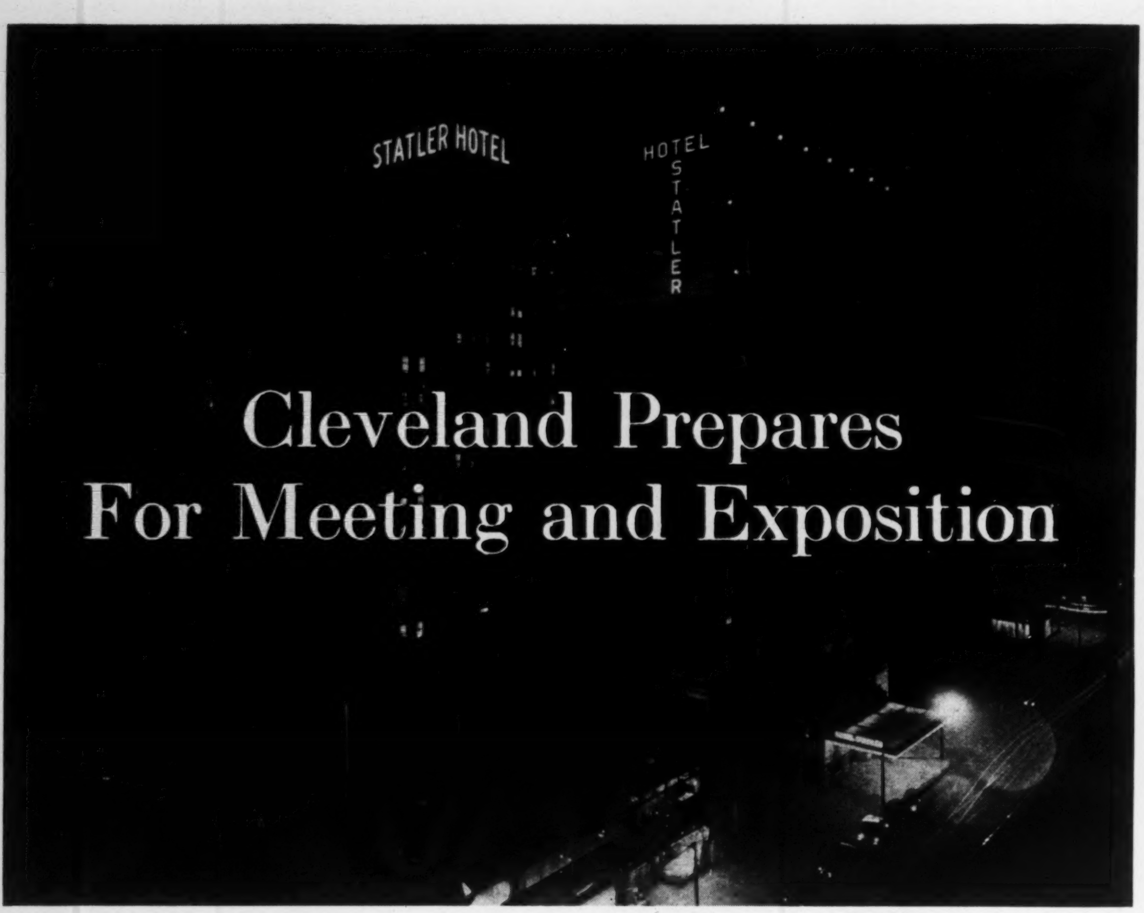
D. E. French
Newark, N. J.

Prof. F. E. Giesecke
Austin, Texas

Prof. A. P. Kratz
Urbana, Ill.

Prof. G. L. Larson
Madison, Wis.

L. A. Harding
Buffalo, N. Y.



Cleveland Prepares For Meeting and Exposition

THERE is every indication that Cleveland will be the capital of the Heating World in January when the Annual Meetings of the A. S. H. V. E. and the *A. S. R. E.* are held contemporaneously with the 2nd International Heating and Ventilating Exposition. The Society will make its headquarters in the Hotel Statler and the refrigerating men will gather at the Hotel Cleveland. Joint meetings of the two Societies will be held in the Little Theater of the Cleveland Auditorium adjacent to the Exposition.

Sessions and social events have been planned by the arrangements committee with the idea of giving members an opportunity of attending the functions of both Societies and also to have ample time to visit the Exposition, where an extensive display of equipment will be shown in operation. The laboratories of manufacturers have brought forth many ingenious devices that will be offered for public inspection for the first time at the Exposition. A representative number of manufacturers of heating, ventilating and refrigerating equipment will have exhibits, so that the success of the undertaking is assured. Already two-thirds of the available exhibit space has been contracted for and every section of the industry will be represented.

The various allied associations will have booths, among which are the A. S. H. V. E., *A. S. R. E.*, the *National Warm Air Heating Association, A. G. A., American Oil Burner Association* and the *Heating & Piping Contractors Association.* With this interest manifest it is expected that this show will surpass the record attendance in 1930 when 57,000 visitors were registered.

On Oct. 7, the joint Committee on Arrangements met at the Hotel Statler, Cleveland, and under the guidance of Messrs. Weager and Keith various entertainment features were discussed. An event of universal appeal will be a trip to Akron to the plant of the Goodyear-Zeppelin Corp. where the dirigible *Akron* is housed. It is hoped that the big ship will be in the hangar at that time.

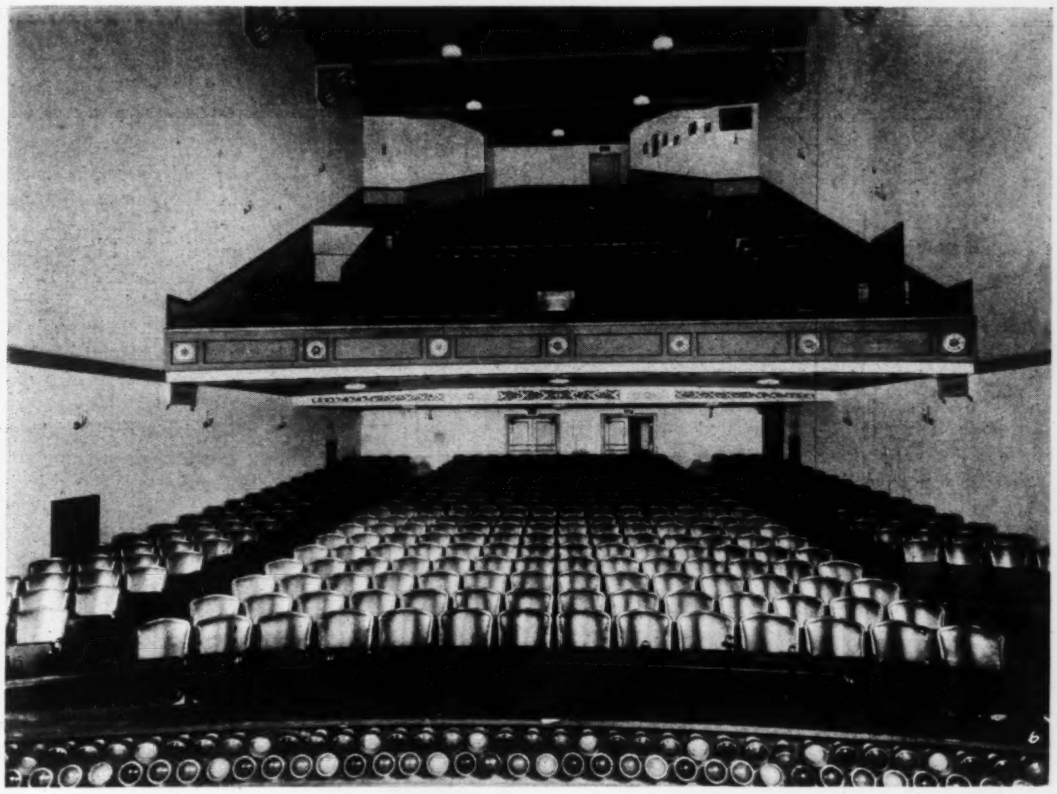
According to the tentative program, Monday, January 25, is reserved for Registration, Committee and Council meetings and a joint dinner for the Officers and Council members of the two Societies.

The opening session is to be held at the Hotel Statler at 10 o'clock Tuesday morning and in addition to various committee reports two technical papers will be given. An entire afternoon has been reserved for a discussion of two important committee reports, namely: Committee on Ventilation Standards and Committee on Revision of Constitution and By-Laws.

On Wednesday and Thursday mornings, joint sessions of the two Societies are to be held in the Little Theater of the Cleveland Auditorium. This room, seating 800, is adjacent to the Exposition. Thursday afternoon the session will be held at the Hotel Statler, as will the final session on Friday morning.

The entertainment program is varied and attractive, each Society providing special features for the ladies and arranging for several joint events. The annual banquet and dance of the *A. S. R. E.* will be held on Wednesday and the A. S. H. V. E. banquet will come on Thursday night so that members interested in both organizations will have an opportunity of attending both functions.

Members should be making plans to come to Cleveland in January. Every encouragement is offered in the way of low hotel rates, reduced railroad fare and a splendid program of technical papers and social gatherings. The opportunity for discussion of engineering subjects, the personal contacts and the varied attractions of the Exposition combine to make this 38th Annual Meeting of the A. S. H. V. E. an outstanding event in the history of the profession and industry.



Little Theater in Cleveland Public Hall

38th ANNUAL MEETING

American Society of Heating and Ventilating Engineers

January 25-29, 1932
Hotel Statler, Cleveland, Ohio

Monday—January 25th

10:00 A. M.—Meeting of the Council
12:30 P. M.—Luncheon for Officers and Authors
2:00 P. M.—Meeting Committee on Research
7:00 P. M.—Joint Dinner-Meeting of the Councils of A. S. H. V. E. and *A. S. R. E.* and their wives (Hotel Cleveland)

Tuesday—January 26th

10:00 A. M.—First Session (Ballroom Hotel Statler)
Address of Welcome
Greeting by Pres. W. H. Carrier
Report of Tellers of Election
Report of Council
Technical Papers (*two*)
Report of the Finance Committee
Report of the Secretary
2:30 P. M.—Report of Committee on Ventilation Standards
Discussion—Revision of Constitution and By-Laws
7:30 P. M.—Inspection of Second Heating and Ventilating Exposition
9:00 P. M.—Informal Reception and Dance for A. S. H. V. E. and *A. S. R. E.* members (Hotel Statler)

Wednesday—January 27th

10:00 A. M.—Third Session (Cleveland Auditorium)
Joint Meeting with *A. S. R. E.*—Program arranged by A. S. H. V. E.—Pres. W. H. Carrier presiding
2:00 P. M.—Trip to Akron (Goodyear-Zeppelin Hangar)
7:00 P. M.—Dinner for Past-Presidents and Wives (Hotel Statler)
8:00 P. M.—Inspection of Second Heating and Ventilating Exposition

Thursday—January 28th

10:00 A. M.—Fourth Session (Cleveland Auditorium)
Joint Session with *A. S. R. E.*—Program arranged by *A. S. R. E.*—Glen Muffly presiding
2:30 P. M.—Fifth Session
Technical Papers (*three*)
7:00 P. M.—Annual Banquet and Dance (Hotel Statler)

Friday—January 29th

10:00 A. M.—Sixth Session (Hotel Statler)
Technical Papers (*two*)
Installation of Officers
Resolutions of Officers
Adjournment
12:30 P. M.—Luncheon and Meeting of Council
2:30 P. M.—Inspection Trip to Cleveland Industrial Plants

COMMITTEES

T. A. Weager, *General Chairman*

Finance: Walter Klie, *chairman;* W. E. Stark and H. M. Nobis.

Banquet: W. C. Kammerer, *chairman;* C. W. St. Clair, Vincent Eaton.

Reception and Registration: C. F. Eveleth, *chairman;* F. A. Kitchen, C. Gottwald.

Transportation and Inspection Trips: F. H. Morris, *chairman;* C. J. Deex, J. E. Beyer.

Entertainment: R. G. Davis, *chairman;* E. H. Pogalies, S. H. Givelber.

Publicity Committee: M. F. Rather, *chairman;* D. E. Humphrey.

A. S. R. E.: D. F. Keith.

Local Chapter Reports

Massachusetts

October 5, 1931. The Massachusetts Chapter of the Society held the first of the fall meetings at 16 Haywood Place, Boston.

In the absence of President Moulton, D. S. Boyden welcomed the 41 members and guests, who had assembled at luncheon, and presented a communication from C. V. Haynes regarding the Research work of the Society. The members present indicated their support of the work being done by the Committee on Research.

J. A. Flemings, Detroit Stoker Co., presented an interesting talk on the different kinds of stokers for heating boilers and their respective merits, and gave data regarding the proper size of combustion chambers.

Pittsburgh

October 5, 1931. The regular monthly meeting of the Pittsburgh Chapter of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was held at the Keystone Athletic Club, with 72 members and guests in attendance. After the usual routine of business was attended to, Pres. W. H. Carrier addressed the members and guests on Servicing the Human Power Plant, which was one of the most interesting and instructive talks presented before the Chapter.

President Carrier brought out the point that a large part of the research had been done at the Society's Research Laboratory in Pittsburgh, under the direction of F. C. Houghten, who is Director. He then explained that buildings are heated, not to heat the human beings in them, but rather to furnish an atmospheric condition that will serve to retard the dissipation of heat from the human body. Air conditioning is merely the supplying of those factors which will cause the human organism to react comfortably, by controlling the temperature and quality of the air. A human being is really a power plant or heat engine, considered physiologically, his food being converted into mechanical energy. By means of slides, President Carrier explained the heat and moisture loss data resulting from temperature, humidity, and air motion studies made in the Laboratory, and also brought out the relation of these data to air conditioning and man's comfort.

He described the different systems used for air distribution and conditioning in general, and compared them. He stated that the problems of ventilating are largely the same as those of air conditioning, and went further in saying that we have definite knowledge today of the ventilation requirements from a thermal basis, but there is still a great field left unexplored. A rising vote of thanks was tendered to President Carrier by the members and guests in appreciation for his enlightening talk.

At the close of the talk, a number of excellent questions were raised from the floor, and full discussion was participated in. T. N. Adlam, a member of the *British Institution of Heating and Ventilating Engineers,* Birmingham, England, who is visiting this country, brought up several interesting points.

C. W. Farrar, Buffalo, who is chairman of the Membership Committee, gave a very brief but interesting talk concerning membership in the Society. In his inimitable way he suggested the slogan "Every member get a member."

Toronto

October 5, 1931. The first regular meeting of the 1931-32 season of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was held at the Prince George Hotel, with an attendance of 24 members.

Following dinner, Pres. H. S. Moore announced the acting chairmen for the various committees for the coming year as follows: E. R. Cauley—*Publicity and Entertainment;* J. S. Wood—*Attendance Committee;* W. R. Blackhall—*Membership Committee.*

The report on the recent meeting held at the City Hall on the question of Smoke Abatement in the downtown section of the city of Toronto, which was presented by M. F. Thomas, was accepted by the Chapter. A. S. Leitch made the motion to accept it and H. J. Church seconded the motion. Mr. Leitch then read a statement of the research activities of the Society in 1931.

President Moore made a special appeal to the members to be present at the next meeting, which is scheduled for November 10, at which Pres. W. H. Carrier will be principal speaker of the evening. The subject of President Carrier's address will be "Economic Viewpoint in Air Conditioning for Human Comfort."

Mr. Church extended a vote of thanks to the Entertainment Committee for the excellent way in which they handled this opening meeting.

Later in the evening the members adjourned to Karry's Bowling Alley where they enjoyed a pleasant evening of bowling.

R. E. Jones Joins Thermo-Service

Raymond E. Jones is now in charge of the Philadelphia territory for Thermo-Service, Inc., with offices in the Lafayette Building, according to a recent announcement. Mr. Jones has an extended acquaintance in the heating field through his many years as representative of the American Radiator Co., Spencer Heater Co., and lately as president of Haynes Selling Co. He has been active in organization affairs and served as President of the Phildelphia Chapter, A. S. H. V. E.

Cleveland Engineering Society Selects New Quarters

The Cleveland Engineering Society, which had been located in Carnegie Hall for many years where it has enjoyed the use of the club rooms, meeting room, exhibition hall and general offices, is now located on the fourth floor of the Hanna Building on Euclid Avenue in Cleveland. The new location will provide the same facilities as the club had in the past with some additions and improvements.

Bituminous Coal Conference November 16-21, 1931

The 3rd International Conference on Bituminous Coal will be held at the Carnegie Institute of Technology, Pittsburgh, November 16 to 21, 1931.

The purpose of the Congress is similar to those held in 1926 and 1928 by the Institute, which were to present for discussion the results of recent studies of coal, emphasizing improved methods of utilization and their economic value.

Among the subjects to be discussed at this Conference are: Coal as locomotive and steamship fuel; coal in the electric and gas industries; competition of coal with other fuels in heating buildings; coal in the metallurgical industries; coal and its by-products in the chemical industries; problems of coking; origin and classification of coals; cleaning and preparation; mechanism of combustion; pulverized coal; relationships of coal, oil, and natural gas; coal versus water power; hydrogenation and liquefaction of coal; fertilizers from coal; economics of low temperature carbonization; storage and weathering of coal; smoke abatement; and stream pollution.

Employment Survey

An employment survey was instituted recently in the interest of A. S. H. V. E. members and the return cards already received indicate an immediate demand for the services of several types of engineers.

Inquiries should be addressed to the Headquarters office of the Society, 51 Madison Ave., New York, N. Y., by those interested in employment opportunities.

Grand Rapids Members Meet

A group of thirty members and prospective members gathered in Grand Rapids on October 8, and voted to petition the Council of the Society for a charter authorizing the organization of a local chapter in that city.

With O. D. Marshall acting as chairman of the meeting, J. R. McColl, Detroit, was introduced and briefly outlined the History of the Society. W. A. Rowe, Detroit, talked on the Current Activities of the Society and J. F. McIntire explained the Benefits of Chapter Membership. The regulations governing chapter organization and operation were given in detail by A. V. Hutchinson, Secretary of the Society, who also told about the policy other chapters followed in arranging programs that would be entertaining and instructive.

The feature of the evening was the address of Pres. W. H. Carrier, who told of the practical value of research in the establishment of standards of ventilation. He gave the assembled members an intimate glimpse of the valuable work being done in the Society's Laboratory for the benefit of the whole profession and the public.

Due to the efforts of W. W. Bradfield, W. J. Carroll and others, the following men were in attendance:

W. W. Bradfield, Grand Rapids, Mich.; W. L. Taze, Grand Rapids, Mich.; C. H. Morton, Grand Rapids, Mich.; Karl L. Ziesse, Grand Rapids, Mich.; Will J. Pulte, Grand Rapids, Mich.; C. Lammers, Grand Rapids, Mich.; P. T. Busch, Grand Rapids, Mich.; O. J. Dykman, Grand Rapids, Mich.; John J. Wieland, Grand Rapids, Mich.; Jos. J. Troske, Grand Rapids, Mich.; J. Stuart Knee, Grand Rapids, Mich.; Chas. H. Alexander, Grand Rapids, Mich.; Wm. A. Rowe, Detroit, Mich.; H. Christenson, Battle Creek, Mich.; J. F. Johnston, Jr., Ferrysburg, Mich.; C. J. Walsh, Ferrysburg, Mich.; E. W. Jaynes, Grand Rapids, Mich.; J. M. Jaynes, Grand Rapids, Mich.; Joseph Van Rossum, Grand Rapids, Mich.; W. J. Carroll, Grand Rapids, Mich.; W. J. Temple, Kalamazoo, Mich.; V. C. Dean, Grand Rapids, Mich.; C. R. McConner, Kalamazoo, Mich.; S. H. Downs, Kalamazoo, Mich.; J. R. McColl, Detroit, Mich.; W. H. Carrier, Newark, N. J.; O. D. Marshall, Grand Rapids, Mich.; J. F. McIntire, Detroit, Mich.; A. V. Hutchinson, New York City.

Indianapolis Members Entertain President Carrier

At a luncheon meeting October 15, Indianapolis members and guests of the Society assembled at the Chamber of Commerce to hear Pres. W. H. Carrier speak on the importance of research in establishing ventilation standards.

J. J. Hayes presided and introduced the President, whose talk was thoroughly enjoyed by those in attendance.

November Council Meeting in Buffalo

A meeting of the Council has been called for November 9 at the Hotel Statler, Buffalo, N. Y., by President W. H. Carrier. Some of the matters to be considered are the Annual Meeting Program, the Report of the Committee on Code for Testing and Rating Unit Ventilators and the Report of the Committee on Journal Contract.

The Western New York Chapter is to meet in the evening at the Hotel Buffalo and the Officers and Council members will be special guests and President Carrier is to be the featured speaker. The title of his talk will be Servicing the Human Power Plant, which deals with the effect of temperature, humidity and air motion on physiological reactions.

On the night of November 10, President Carrier will speak at the November meeting of the Ontario Chapter in Toronto.

Members to Nominate Candidates for Anderson Medal

THE Committee on Award of the F. Paul Anderson Medal extends an invitation to all members of the American Society of Heating and Ventilating Engineers, to submit the names of candidates for the first award of the F. Paul Anderson Medal, to be made at the 38th Annual Meeting of the Society January, 1932.

The conditions are briefly as follows:

1. Any member of the American Society of Heating and Ventilating Engineers in good standing during 1931 is eligible for nomination..
2. Each nomination sent to the Committee on Award must be accompanied by a statement of the candidates accomplishments specifically mentioning the work done or services performed and indicating in what way such work or services have been outstanding in the field of heating, ventilating or air conditioning, during any year or years prior to 1931.
3. All nominations must be in the hands of the Committee on *November* 30, 1931, and are to be addressed: Committee on Award of the F. Paul Anderson Medal, c/o Secretary, American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York, N. Y.

The personnel of the Committee on Award is as follows: Prof. A. C. Willard, Urbana, Chairman; Thornton Lewis, Newark, N. J.; L. A. Harding, Buffalo; F. B. Rowley, Minneapolis; W. T. Jones, Boston and F. C. McIntosh, Pittsburgh.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 8 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by November 10,1931, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Fox, Otto, Vice-Pres., The Bryant Heater & Mfg. Co., Cleveland, Ohio	W. E. Stark J. E. Kinner	L. C. Harvey
Humphreys, Clark M., Carnegie Institute of Technology, Pittsburgh, Penna.	F. C. Houghten J. L. Blackshaw	C. W. Wheeler H. E. Digby
Rydell, Carl A., Carrier-Lyle Corp., Buffalo, N. Y. (*Advancement*)	V. S. Day G. G. Schmidt	
Schweikert, John N., Vice-Pres., The Bryant Heater & Mfg. Co., Cleveland, Ohio	W. E. Stark J. E. Kinner	L. C. Harvey
Slayter, Games, Pres., Games Slayter, Inc., Cons. Engrs., Detroit, Mich.	W. A. Rowe J. D. Hoffman	M. S. Kice, Jr. H. E. Paetz
Tarr, Harold M., 21 Montgomery St., Arlington Heights, Mass.	A. Gale Schanze F. P. Gilmore	Leslie Clough D. S. Boyden
Temple, Walter J., J. A. Temple Co., Kalamazoo, Mich.	D. L. Taze J. J. Troske	W. W. Bradfield C. H. Morton
Vogelbach, Oscar, Chief Engineer, Guilbert & Betelle, Archts., Newark, N. J.	E. A. Vivarttas A. Pieron—*Non-Member*	M. G. Hook G. G. Schmidt

Candidates Elected

In the past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

Ourusoff, Leon, Washington Gas Light Co., Washington, D. C.

Taze, Donovan Long, Dist. Mgr., American Blower Corp., Grand Rapids, Mich.

Weiner, Raymond Louis, Gibbs & Hill, New York, N. Y.

Winterbottom, Ralph F., Engr., Winterbottom Supply Co., Waterloo, Iowa. (*Reinstatement*)

ASSOCIATES

Busch, Philip J., Salesman, American Radiator Co., Detroit, Mich.

Dykman, Orrie John, Heating & Plumbing Contractor, 818-24 Lake Michigan Dr., Grand Rapids, Mich.

Finch, Stanley B., Supervisor, Architects and Builders Division, Brooklyn Union Gas Co., Brooklyn, N. Y.

Lammers, Cornelius, Superintendent, Glendon A. Richards Co., Grand Rapids, Mich.

Marshall, Orville D., Sales Engr., Illinois Engrg. Co., Chicago, Ill.

Morton, Charles H., 524 Murray Bldg., Grand Rapids, Mich.

Troske, Joseph John, Vice-Pres. & Genl. Mgr., Vander Waals-Troske Co., Inc., Grand Rapids, Mich.

Heating · Piping and Air Conditioning

EDITORIALS

Excessive Competition

What is excessive competition? It would be interesting to find out what it is and how it operates in the fields of heating, piping and air conditioning if it be true, as at least one economist maintains, that the business depression is due to excessive competition. He cites an example:

A manufacturer sees his competitor forging ahead. He decides that he has run long enough on the conservative policy of quality, adequate price and consistent salesmanship.

He says, "We must increase our sales fifty per cent this year." He puts it up to the salesmen, "If you cannot sell your quota do not come back home." Quality is cut to make an attractive price and to undersell others. Next year the same story. Another increase of fifty per cent in production and sales. The stuff must be sold. To the salesmen, "Get business honestly if you can, but get business." The sales manager pins his faith to high-pressure selling and high-pressure advertising. "Clever salesmanship and advertising," he says, "can sell doorknobs in a community of tents." He may preach honesty, but the salesmen know he means "Sell, sell, sell. Truth, quality, price—juggle them as you will, but sell."

Any high-school boy should be able to figure out the probable result. Sooner or later the buyer will protest. He will set up barriers against such salesmanship and such advertising. He will refuse to buy. When a sufficient number of buyers become of such a mind, there will be a business depression, and the great majority of honest manufacturers suffer along with the few who started it all. Such a competitor is like a reckless automobile driver, a menace not only to himself but to everyone else on the road.

There are examples of excessive competition in the heating, piping and air conditioning fields. In buying many types of heating, piping and air conditioning equipment, the first cost is not always given its proper relative importance.

For instance, more important than first cost of pipe insulation is the quality of the insulation and the quantity applied. For if it be lacking in quality or quantity the buyer pays not only once but many times and year after year. He pays in increased fuel bills, reduced capacity and increased labor and maintenance costs.

And it is not only the owner who pays. The architect pays, the consulting engineer pays, the plant engineer pays, the stockholders pay and the public pays because first cost was permitted to overshadow other and greater considerations. It may be the net saving per year and not the saving in the first cost that is important.

An eminent engineer said recently that more economies can be effected by correct pipe insulation than by any other detail of construction. But too often the amount that may be saved or the type or amount of insulation which should be used is arrived at by guessing. It may be just as bad to use too much as too little. The critical point between economy and waste readily may be determined, however, by the proper use of one-tenth of a cent's worth of graphite from the end of a good lead pencil properly applied to a sheet of paper by a competent and conscientious engineer.

Proper sales engineering—backed by clear and conscientious advertising—will do much to develop a confidence and maintain a prestige which will weather the storms of business depressions, while those who indulge in excessive competition and are partly responsible for the storms become its victims.

"Professor of Economy"

By revising the lay-out of its air piping systems, a large mine obtained important savings in the cost of maintenance and the cost of power. In an art museum, valuable exhibits are prevented from deteriorating by the introduction of air maintained at the proper humidity. An industrial plant supplies high, intermediate and low pressure steam for testing, process and heating by means of piping systems designed to accomplish the desired purposes economically. Through careful study of the problem, a power plant is assured that the flanged joints in its high temperature steam lines will have long life.

These are but a part of the "economy articles" which are published in this issue. They will point the way for many engineers to make savings in their own plants and structures. Every article and advertisement has a personal significance to the careful reader. Experience is a wonderful teacher, but the printed word is a lot less expensive and represents the experience of many. It should be used freely.

Survival of the Fittest

At a recent joint meeting of the *American Society for Testing Materials* and the *Western Society of Engineers,* Frank D. Chase, president of the *W. S. E.,* said in part:

"This nation faces today, as never before, a period of intense industrial competition, both at home and abroad. It is a time of the 'survival of the fittest'—and there is only one way in which we can survive, either as individuals or as a nation, and that is by the introduction and practice of economies, in materials, methods and processes, to secure reductions in manufacturing or production costs." Mr. Chase went on to stress the role the *A. S. T. M.* plays in the field of economics in industry.

The correct application of engineering principles to the design, the installation and the maintenance of heating, piping and air conditioning systems in industrial and commercial buildings and other properties is a necessary adjunct to the attainment of minimum production costs. It is the duty of the engineer to his client or employer to recommend the purchase of that equipment which will promote efficiency, to see that it is installed in a manner which will enable its proper function to be fulfilled, and to maintain it so that its full effectiveness can be realized economically.

Air Conditioning Plays an Important Part in Our Lives

The powerful effect of air environment on the human body and mind is becoming more and more widely recognized. In industrial plants, comfortable atmospheric conditions play no small part in improving the efficiency and health of the workers; in offices, mistakes and "let-downs" are minimized by air of the proper temperature, humidity, and air motion; in schools, growth and development of the children is dependent to some extent on the maintenance of air of the necessary purity and physical characteristics; more and more hospitals are providing cooled air, particularly to protect infants from excessive heat of summer, in addition to providing filtered and conditioned air the year 'round.

As a business proposition, the necessity of air conditioning for theaters, restaurants, hotels, etc., is already well recognized, and air conditioning for the benefit of materials in process is so essential that its application is wide.

It is encouraging to note the increasing attention being given to air conditioning for health and comfort, such as that accorded it by the committee on growth and development, White House conference on child health and protection.

Air conditioning has an important role to play—one that is growing in importance as time goes on.

Practical Piping Problems

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Installing Industrial Air Piping

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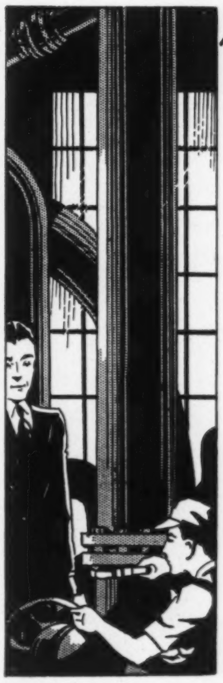

W. H. Wilson's Monthly Discussion of the Problems He Meets From Day-to-Day

Proper installation of the pipe lines for industrial use requires careful consideration in order that the best results may be obtained at reasonable installation cost, with economy in operation, and low maintenance and upkeep costs.

One thing depends upon another, and local conditions determine what material should be used and how it should be installed. Designing engineers, installation men, operating and maintenance employes should all be considered in any installation of industrial equipment, particularly pipe lines.

The principal complaints that the maintenance department usually receives regarding compressed air pipe lines refer to moisture and water in the lines, and to accumulations of rust, pipe scale and sediment.

Water and moisture in exposed air piping will cause trouble in freezing weather. Air containing moisture may cause serious trouble at any time in the operation of paint spraying equipment, air operated tools, and sand blasting and cleaning equipment.

Use of Separators

Many devices are used for eliminating this moisture. Some plants provide separators on the main and branch lines, pockets and water legs at various places, and drain valves at the foot of riser pipes, though air pressure may be lost by frequent blowing out of the drain valves on large separators, pockets and risers. Good results are also obtained by placing suitable separators or moisture eliminators, of a smaller size, as near as is practically possible to the individual tools, paint spray booths or other equipment where dry air is required. Separators at various locations mean maintenance work in the way of cleaning and keeping them in proper working order, but the extra amount of work required will be repaid in the efficiency obtained by the use of dry and clean air.

Containers filled with charcoal are sometimes used to a good advantage. The moisture is absorbed by the charcoal and dry air passes from the container through the outlet pipe.

Air compressors equipped with after-coolers to cool the air as it leaves the compressor will assist in keeping moisture from entering the pipe lines. A suitable method of draining should be provided for removing the moisture from the after-cooler.

Anti-sweat Insulation on Intake

The air supply to the compressor is usually from the outside atmosphere and the intake pipe, when passing through a warm room to the compressor, is apt to sweat when cold air from the outside is being drawn in. Suitable anti-sweat insulation should be applied to the intake pipe in order that the dripping moisture will not be objectionable.

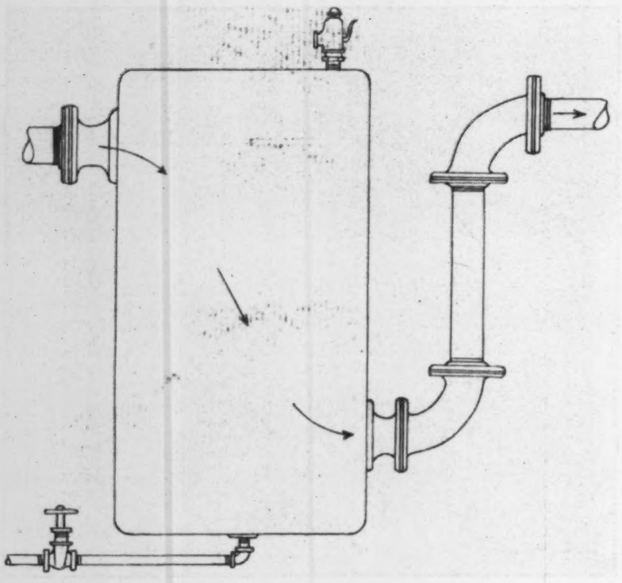

Fig. 1—Piping Connections for an Air Receiver

Alcohol Clears Line

A pipe line transmitting compressed air which is not free from moisture, and installed underground in contact with the earth, if not equipped with insulation or other protection, will collect moisture and water. If the line is located near the surface of the ground, trouble will be experienced in freezing weather. In case of stoppage in such a line by ice, a small amount of alcohol injected in the line will frequently clear it.

Piping the Receiver

The air discharge pipe from the compressor is usually connected to a tank or receiver, which acts as a storage reservoir to steady the air pressure in the supply lines when the demand is intermittent, causing the compressor to work steadily.

This receiver can be used for separating some of the moisture from the air. It should be placed where it will remain cold or as cool as possible. Out-of-doors and on the north side of a building is usually a good location. Warm air has a tendency to rise and also to hold its moisture; therefore, if the discharge pipe of the air compressor be connected near the top of the receiver the air leaving the receiver to supply the shop air piping will come from the lower part of the receiver, and will be the cooler air. The air in cooling has a tendency to drop some of its moisture. The receiver should have a suitable drain provided with a reliable trap to remove the moisture. The air outlet pipe for supplying the air to the shops from this receiver should be arranged so as not to interfere with the drain. Figs. 1 and 2 show different ways of arranging the piping connections to air storage receivers.

Trouble from Moisture in an Air Line

On account of an emergency it was necessary to operate a steam driven, reciprocating, duplex oil pump with compressed air, at a pressure of 80 lb. per sq. in. A small pipe connection was taken from a 6-in. air main located a short distance away in a concrete tunnel. This was in September, the outside temperature during the day was from 50 to 70 F and there was rain most of the time.

The temperature of the tunnel was approximately 80 F. The air compressor was located 1,500 ft. from the pump. The pump was arranged to exhaust through a short length of pipe to the atmosphere. After the pump had been in operation a short time the exhaust pipe was covered with frost, trouble was experienced by the pipe clogging several times with ice and frequently particles of ice were blown out with the exhaust air.

Consider These Points

The following points should be considered before installing piping for compressed air:

Provide for the removal of the moisture and a supply of clean air. This includes a clean supply from the atmosphere, by using filters, if necessary. The location of drains, selection and location of suitable strainers, moisture separators and traps for drainage should be studied carefully.

Special attention should be given to the selection of traps, when used to drain the water and moisture from compressed air lines and receivers automatically. Some traps that work well on steam may not be suitable for use on compressed air equipment. Special attention should be given to the installation and venting of the trap so that it will not become air bound.

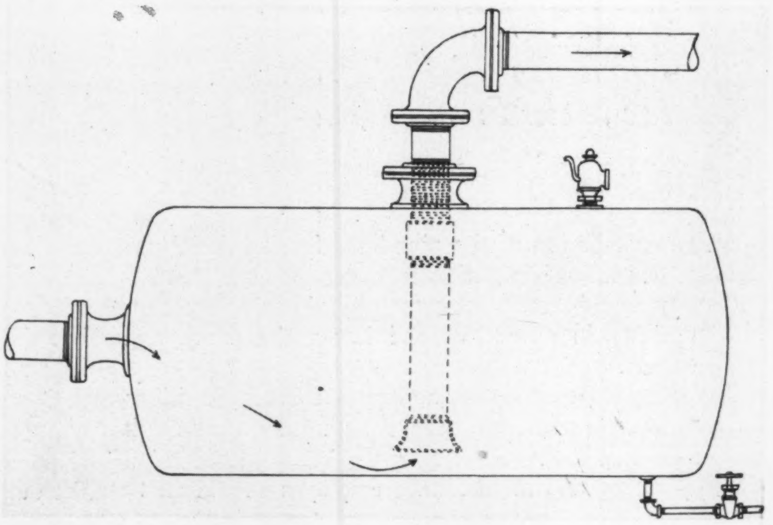

Fig. 2—Piping for an Air Receiver

When screwed pipe joints are assembled a good grade of shellac applied to the threads before making up the joint will aid materially.

Use a good grade of pipe.

Install pipe lines that are large enough so as to get the maximum supply and pressure desired, with a slow air flow. There is no real objection to an air line being too large, excepting the initial cost. There are no condensation losses to increase the cost of upkeep as there would be in a steam main that was too large. The extra volume of air required to fill the line when starting up will be repaid by the air moving slower, which will, to some extent, prevent its carrying moisture, lubricating oil and pipe scale. The volume of air in a large line acts to steady pressure when demand is irregular.

Service connections from the branch lines should preferably be taken from the top of the pipe line, when this line is in a horizontal position, as this lessens the chances of dirt and scale getting into the service connections. If there is any entrained water in the line it will not be likely to flow into the service drops. Arrangements must be made to remove the water from the lines by suitable drains and traps at the low points.

Development of Clavarino Formula

By H. A. Wagner*

THE diagrams given in "Curves Aid in Design of Thick-Walled Tubes and Cylinders," by F. E. Wertheim,[1] permit the ready solution of the somewhat cumbersome formula of Clavarino for determining wall thickness of cylinders or tubes. The example given on the second page of the article could be interpreted as the application of the curves of Fig. 1 to the problem of determining the wall thickness of a hydraulic cylinder when the internal area or diameter of cylinder required to perform a certain function is known. Likewise, in determining the wall thickness of pipes, where the outside diameter is arbitrarily set due to commercial practice, the curves of Fig. 2 are applicable.

It may be of interest to those who are accustomed to the use of the conventional form of Clavarino's equation to follow the derivation of the form employed by Mr. Wertheim from the general equation, which is:

$$f = p \frac{d^2 (1 - 2\lambda) + D^2 (1 + \lambda)}{D^2 - d^2} \qquad (1)$$

where f = fiber stress at inner surface of wall, lb. per sq. in.

p = internal unit pressure, lb. per sq. in.

d = inside diameter of cylinder, in.

D = outside diameter of cylinder, in.

λ = Poisson's ratio.

If Poisson's ratio is assumed as 0.3, the above formula reduces to the following, which is that generally encountered:

$$f = p \frac{1.3D^2 + 0.4 d^2}{D^2 - d^2} \qquad (2)$$

If Poisson's ratio is assumed as 1/3, the following equation results:

$$f = p \frac{4/3 D^2 + 1/3 d^2}{D^2 - d^2} \qquad (3)$$

This may be transformed by the following steps into the form used in Mr. Wertheim's article:

$$f D^2 - f d^2 = 4/3 D^2 p + 1/3 d^2 p \qquad (4)$$

$$D^2 (f - 4/3p) = d^2 (f + 1/3p) \qquad (5)$$

$$\frac{D^2}{d^2} = \frac{f + 1/3p}{f - 4/3p} = \frac{3f + p}{3f - 4p} \qquad (6)$$

$$\frac{D}{d} = \left(\frac{3f + p}{3f - 4p}\right)^{1/2} \qquad (7)$$

$$D = d + 2t \qquad (8)$$

$$d + 2t = d\left(\frac{3f + p}{3f - 4p}\right)^{1/2} \qquad (9)$$

$$2t = d\left(\frac{3f + p}{3f - 4p}\right)^{1/2} - d \qquad (10)$$

$$t = \frac{d}{2}\left[\left(\frac{3f + p}{3f - 4p}\right)^{1/2} - 1\right] \qquad (11)$$

$$t = r \times C, \text{ where } C = \left(\frac{3f + p}{3f - 4p}\right)^{1/2} - 1 \qquad (12)$$

The formula involving the outside diameter may be obtained by replacing d by $D - 2t$ in Equation (7) and completing the solution as above.

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* Engineer, The Detroit Edison Company, Detroit, Mich.

[1] Heating, Piping and Air Conditioning, October, 1931, p. 850.

Conventions and Expositions

National Association of Practical Refrigerating Engineers: Annual convention and exposition, Nov. 10-13; Municipal Auditorium, Houston, Texas. Secretary, Edward H. Fox, 435 N. Waller Ave., Chicago.

International Acetylene Association: Annual convention Nov. 11-13; Congress Hotel, Chicago. Secretary-Treasurer, A. Cressy Morrison, 30 E. 42nd st., New York City.

Third International Conference on Bituminous Coal: November 16-21; Carnegie Institute of Technology, Pittsburgh, Pa. Chairman, Thomas S. Baker, President, Carnegie Institute of Technology.

American Society of Mechanical Engineers: Annual meeting, Nov. 30-Dec. 4; New York City. Secretary, Calvin W. Rice, 33 W. 39th st., New York City.

Welding Conference: Annual conference to be held by engineering extension department, Purdue University, Lafayette, Ind., Dec. 10-11. In charge, Prof. W. A. Knapp, Purdue University, Lafayette, Ind.

Society of Rheology: Annual meeting, Dec. 28-30; Rochester, New York. Chairman of Program Committee, Eugene C. Bingham, Gayley Chemical Laboratory, Lafayette College, Easton, Pa.

American Society of Heating and Ventilating Engineers: Annual meeting, Jan. 25-29; Cleveland, Ohio. (Joint meeting with A. S. R. E.) Secretary, A. V. Hutchinson, ~~51 Madison ave., New York City~~.

American Society of Refrigerating Engineers: Annual convention, Jan. 26-29 (Joint meeting with A. S. H. V. E.); Hotel Cleveland, Cleveland, Ohio. Secretary, David L. Fiske, 37 W. 39th St., New York City.

International Heating and Ventilating Exposition: Jan. 25-29; Cleveland Auditorium, Cleveland, Ohio. Manager, Charles F. Roth, International Exposition Co., Grand Central Palace, New York City.